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A Complete System
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in RAILWAY TRANSPORTATION

Ten Volumes
Profusely Illustrated

Calvin F. Swingle
EDITOR IN CHIEF

ASSISTED BY A CORPS OF
MECHANICAL EXPERTS

PUBLISHERS
NATIONAL INSTITUTE of PRACTICAL MECHANICS
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LOCOMOTIVE RUNNING AND MANAGEMENT

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COVERING THE USE OF

INJECTORS, LUBRICATORS AND THE SETTING OF VALVES

ALSO A COMPLETE DESCRIPTION

OF

WALSCHAERT AND OTHER VALVE GEARS

THEIR DEFECTS AND HOW TO REMEDY THEM

VOL. II

ILLUSTRATED

BY

CALVIN F. SWINGLE, M. E.

AND A CORPS OF PRACTICAL MECHANICAL EXPERTS

PUBLISHERS

NATIONAL INSTITUTE OF PRACTICAL MECHANICS

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LOCOMOTIVE RUNNING AND MANAGEMENT.

A man, in order to be a successful locomotive engineer, should understand not only the running of the engine over the assigned route, in accordance with train rules, and regulations, but he should also thoroughly understand the management of the machine.

He should be able to manage and care for his engine, while on the trip, in such manner as to get the largest amount of work performed with the least expenditure of fuel and oil, and the least wear and tear on the machine.

It is one thing to simply run a locomotive, but the running and management of a locomotive in the proper way means still more, and it is the purpose of the author, in these pages, to endeavor to help the young man aspiring to become a successful engineer, and at the same time to present some hints and suggestions that may prove to be helpful to old runners.

It certainly is a truthful saying that a man never becomes too old to learn, and especially is this the case with engine men when we consider the many changes, and improvements that are continually being made on locomotives.

The care and operation of locomotive boilers is thoroughly discussed in Vol. 1.

The use of the injector, the proper feeding of the water to the boiler, the handling of the fire, and the making of the steam and the method of its passage from the dome to the steam chests have all been carefully treated upon.

Before proceeding to a study of the mechanism of the

locomotive, the author deems it fitting to devote a space to the consideration of the nature and composition of two very important factors in the running and management of a locomotive, viz.: water and fuels.

WATER.

Water. The elements that enter into the composition of pure water are the two gases, hydrogen and oxygen, in the following proportions:

By volume, hydrogen 2, oxygen 1.
By weight, hydrogen 11.1, oxygen 88.9.

Water contracts and becomes denser in cooling until it reaches a temperature of 39.1° F., its point of greatest density. Below this temperature it expands and at 32° F. it becomes solid or freezes.

Expansion of Water by Heat. Water expands as the temperature rises above the temperature of maximum density, 39.1° Fahr., it also expands in nearly an equal ratio, as the temperature falls below this point, down to the freezing point.

The volume of pure water at 62° in cubic inches × .0036 = capacity in gallons.

The pressure of a column of water 1 foot high $= \frac{1.000}{2.309}$ = .433 lb., or 6.928 ounces, per square inch pressure.

The pressure of a column of water 33.947 feet, or say 34 feet high = the pressure of the atmosphere, or 14.7 lbs.

The pressure of a column of water $\frac{33.947 \text{ feet high}}{14.7 \text{ lbs.}} =$ 2.309 feet high, or say 2.31 feet high = 1 lb. per square inch pressure.

Table 24.—Temperature and Volume of Water.

Temperature				
At	40°	Fahrenheit,	its volume	1.00000
"	55°	"	"	1.00054
"	65°	"	"	1.00137
"	75°	"	"	1.00255
"	85°	"	"	1.00404
"	95°	"	"	1.00583
"	105°	"	"	1.00800
"	115°	"	"	1.01121
"	125°	"	"	1.01275
"	135°	"	"	1.01551
"	145°	"	"	1.01847
"	155°	"	"	1.02162
"	165°	"	"	1.02500
"	175°	"	"	1.02845
"	185°	"	"	1.03209
"	195°	"	"	1.03590
"	205°	"	"	1.03984
"	212°	"	"	1.04600

One Gallon of Pure Water at 62° Fahr. weighs 10 lbs., and its volume is 277.123 cubic inches, or .160372 cubic foot.

The Volume of 1 lb. of Pure Water—

At 32° Fahr. = 27.684 cubic inches.
At 39.1° Fahr. = 27.680 cubic inches.
At 62° Fahr. = 27.712 cubic inches.
At 212° Fahr. = 28.978 cubic inches.

The volume of one ton of pure water = 35.9 cubic feet.

Water is 815 times heavier than atmospheric air. The weight of a cubic foot of water at 39.1° is approximately

62.5 lbs., although authorities differ on this matter, some of them placing it at 62.379 lbs., and others at 62.425 lbs. per cubic foot. As its temperature increases its weight per cubic foot decreases until at 212° F. one cubic foot weighs 59.76 lbs.

The Weight of a cubic foot of Ice is from 57.5 to 58 lbs., its specific gravity being about .922. Water in freezing expands about 1-12th of its original volume as water, its expansive force at the moment of freezing being estimated at about 32,000 lbs. per square inch. A cubic foot of ice floating in water has about 11-12ths of its volume under water, and 1-12th of its volume above water, and a square foot of ice of any thickness requires a weight equal to 1-11th of its weight to sink it to the surface of the water. The compressive strength of hard pure block-ice is about 20 tons per square foot. The specific heat of ice is .504.

Snow weighs 6 lbs. per cubic foot when freshly fallen, and averages 12 lbs. per cubic foot when moderately saturated with rain. Snow has 12 times the bulk of water, and its specific gravity is .084.

The Standard Measures of water are as follows:—

The weight of one cubic inch of pure water—

At	62°	Fahr. =	252.595	grains.
At	32°	Fahr. =	.03612	lb.
At	39.1°	Fahr. =	.036125	lb.
At	62°	Fahr. =	.03608	lb.
At	212°	Fahr. =	.03451	lb.

The Weight of one cylindrical inch of pure Water at 62° Fahr. = .4533 ounce or .02883 lb.

The Weight of one cubic foot of pure Water—

At 32° Fahr. = 62.418 lbs.
At 39.1° Fahr. = 62.425 lbs.
At 62° Fahr. = 62.355 lbs.
At 212° Fahr. = 59.640 lbs.

The weight of a cubic foot of water is usually taken at 62.4 lbs. per cubic foot for ordinary calculations.

A cubic foot of water at the temperature of maximum density weighs 998.8 ounces—it is usually taken at 1,000 ounces.

Table 25.—Weight of a Cubic Foot of Water from 32° F. to 420° F. Advancing by 10°.

Temperature.	Weight per Cubic Foot.	Temperature.	Weight per Cubic Foot.	Temperature.	Weight per Cubic Foot.
Deg. Fahr.	Pounds.	Deg. Fahr.	Pounds.	Deg. Fahr.	Pounds.
32	62.42	132	61.52	230	59.37
42	62.42	142	61.34	240	59.10
52	62.40	152	61.14	250	58.85
62	62.36	162	60.94	260	58.52
72	62.30	172	60.73	270	58.21
82	62.21	182	60.50	300	57.26
92	62.11	192	60.27	330	56.24
102	62.00	202	60.02	360	55.16
112	61.86	212	59.76	390	54.03
122	61.70	220	59.64	420	52.86

The weight of fresh water compared with that of sea water is as 1 to 1.026, or as 39 to 40.

The mean specific gravity of sea water is 1.026.

The composition of average sea water is as follows:—

Water	96.6	parts.
Chloride of Sodium	2.6	parts.
Chloride of Magnesia	.4	parts.
Sulphate of Soda	.37	parts.
Carbonate of Lime	.02	parts.
Sulphate of Lime	.01	parts.
	100.00	parts.

The ice of sea water contains no salt, because water in freezing parts with all its impurities.

Ordinary sea water contains 1-33 part of its weight of salt, termed one degree of saltness.

Table 26.—Weight of Water from 32° F. to 212° F., and Heat Units per Pound Reckoned Above 32°.—Kent.

Temp., Deg. Fahr.	Weight, lbs. per Cu. Ft.	Heat-units.	Temp., Deg. Fahr.	Weight, lbs. per Cu. Ft.	Heat-units.
32	62.42	0.	48	62.41	16.
33	62.42	1.	49	62.41	17.
34	62.42	2.	50	62.41	18.
35	62.42	3.	51	62.41	19.
36	62.42	4.	52	62.40	20.
37	62.42	5.	53	62.40	21.01
38	62.42	6.	54	62.40	22.01
39	62.42	7.	55	62.39	23.01
40	62.42	8.	56	62.39	24.01
41	62.42	9.	57	62.39	25.01
42	62.42	10.	58	62.38	26.01
43	62.42	11.	59	62.38	27.01
44	62.42	12.	60	62.37	28.01
45	62.42	13.	61	62.37	29.01
46	62.42	14.	62	62.36	30.01
47	62.42	15.	63	62.36	31.01

TABLE 26—Continued.

Temp., Deg. Fahr.	Weight, lbs. per Cu. Ft.	Heat-units.	Temp,, Deg. Fahr.	Weight, lbs. per Cu. Ft.	Heat-units.
64	62.35	32.01	101	62.01	69.08
65	62.34	33.01	102	62.00	70.09
66	62.34	34.02	103	61.99	71.09
67	62.33	35.02	104	61.97	72.09
68	62.33	36.02	105	61.96	73.10
69	62.32	37.02	106	61.95	74.10
70	62.31	38.02	107	61.93	75.10
71	62.31	39.02	108	61.92	76.10
72	62.30	40.02	109	61.91	77.11
73	62.29	41.02	110	61.89	78.11
74	62.28	42.03	111	61.88	79.11
75	62.28	43.03	112	61.86	80.12
76	62.27	44.03	113	61.85	81.12
77	62.26	45.03	114	61.83	82.13
78	62.25	46.03	115	61.82	83.13
79	62.24	47.03	116	61.80	84.13
80	62.23	48.04	117	61.78	85.14
81	62.22	49.04	118	61.77	86.14
82	62.21	50.04	119	61.75	87.15
83	62.20	51.04	120	61.74	88.15
84	62.19	52.04	121	61.72	89.15
85	62.18	53.05	122	61.70	90.16
86	62.17	54.05	123	61.68	91.16
87	62.16	55.05	124	61.67	92.17
88	62.15	56.05	125	61.65	93.17
89	62.14	57.05	126	61.63	94.17
90	62.13	58.06	127	61.61	95.18
91	62.12	59.06	128	61.60	96.18
92	62.11	60.06	129	61.58	97.19
93	62.10	61.06	130	61.56	98.19
94	62.09	62.06	131	61.54	99.20
95	62.08	63.07	132	61.52	100.20
96	62.07	64.07	133	61.51	101.21
97	62.06	65.07	134	61.49	102.21
98	62.05	66.07	135	61.47	103.22
99	62.03	67.08	136	61.45	104.22
100	62.02	68.08	137	61.43	105.23

TABLE 26—Continued.

Temp., Deg. Fahr.	Weight, lbs. per Cu. Ft.	Heat-units.	Temp., Deg. Fahr.	Weight, lbs. per Cu. Ft.	Heat-units.
138	61.41	106.23	176	60.64	144.51
139	61.39	107.24	177	60.62	145.52
140	61.37	108.25	178	60.59	146.52
141	61.36	109.25	179	60.57	147.53
142	61.34	110.26	180	60.55	148.54
143	61.32	111.26	181	60.53	149.55
144	61.30	112.27	182	60.50	150.56
145	61.28	113.28	183	60.48	151.57
146	61.26	114.28	184	60.46	152.58
147	61.24	115.29	185	60.44	153.59
148	61.22	116.29	186	60.41	154.60
149	61.20	117.30	187	60.39	155.61
150	61.18	118.31	188	60.37	156.62
151	61.16	119.31	189	60.34	157.63
152	61.14	120.32	190	60.32	158.64
153	61.12	121.33	191	60.29	159.65
154	61.10	122.33	192	60.27	160.67
155	61.08	123.34	193	60.25	161.68
156	61.06	124.35	194	60.22	162.69
157	61.04	125.35	195	60.20	163.70
158	61.02	126.36	196	60.17	164.71
159	61.00	127.37	197	60.15	165.72
160	60.98	128.37	198	60.12	166.73
161	60.96	129.38	199	60.10	167.74
162	60.94	130.39	200	60.07	168.75
163	60.92	131.40	201	60.05	169.77
164	60.90	132.41	202	60.02	170.78
165	60.87	133.41	203	60.00	171.79
166	60.85	134.42	204	59.97	172.80
167	60.83	135.43	205	59.95	173.81
168	60.81	136.44	206	59.92	174.83
169	60.79	137.45	207	59.89	175.84
170	60.77	138.45	208	59.87	176.85
171	60.75	139.46	209	59.84	177.86
172	60.73	140.47	210	59.82	178.87
173	60.70	141.48	211	59.79	179.89
174	60.68	142.49	212	59.76	180 90
175	60.66	143.50			

The Weight of Salt in a Gallon of Sea-Water may be ascertained as follows: If thirty-three pounds weight of sea-water be evaporated, it will leave one lb. of salt or 1-33 rd of its weight. One gallon of sea-water weighs 10¼ lbs., 1-33 rd part of which is salt, therefore the quantity of salt contained in one gallon is

$$\frac{10.25 \text{ lbs.} \times 16 \text{ ounces}}{33} =$$

4.97 ounces, or in round numbers 5 ounces of salt per gallon of sea-water.

Sea-water Boils at 213.2° Fahr. under the pressure of one atmosphere, or when the mercury in the weather-barometer stands at 30 inches. The boiling point of sea-water varies wth the quantity of salt held in solution, and rises in proportion to its concentration as brine; it also varies with the rise and fall of the weather-barometer.

Saturated Brine boils under the pressure of one atmosphere at 226.4° Fahr.

The point of Saturation of ordinary Sea-water is 12-32 of its weight, when the water is so full of salt that it will hold no more, and it is therefore rapidly precipitated.

The Boiling Point of Sea-water may be calculated from its density as follows: It is found that 1-33rd part of salt increases the boiling point to the extent of 1.2° Fahr.: the boiling point of fresh water being 212° Fahr., that of ordinary sea-water will equal 212°+1.2°=213.2° Fahr. at atmospheric pressure, or when the weather-barometer stands at 30 inches. The following Table has been calculated in this way, each degree of salt representing 5 ounces of salt per gallon of sea-water.

Table 27.—Boiling Points of Sea-Water of Various Densities under one Atmosphere of Pressure When the Barometer Stands at 30 Inches.

Part of Salt.	Degrees of Salt	Ounces of Salt per Gallon.	Boiling Point in Degrees Fahr.
$\frac{1}{33}$	or 1 degree of Salt or	5	213.2
$\frac{3}{66}$	" $1\frac{1}{2}$ " "	$7\frac{1}{2}$	213.9
$\frac{2}{33}$	" 2 " "	10	214.4
$\frac{5}{66}$	" $2\frac{1}{2}$ " "	$12\frac{1}{2}$	215.1
$\frac{3}{33}$	" 3 " "	15	215.6
$\frac{7}{66}$	" $3\frac{1}{2}$ " "	$17\frac{1}{2}$	216.3
$\frac{4}{33}$	" 4 " "	20	216.8
$\frac{9}{66}$	" $4\frac{1}{2}$ " "	$22\frac{1}{2}$	217.5
$\frac{5}{33}$	" 5 " "	25	218.
$\frac{6}{33}$	" 6 " "	30	219.2
$\frac{7}{33}$	" 7 " "	35	220.4
$\frac{8}{33}$	" 8 " "	40	221.6
$\frac{9}{33}$	" 9 " "	45	222.8
$\frac{10}{33}$	" 10 " "	50	224.
$\frac{11}{33}$	" 11 " "	55	225.2
$\frac{12}{33}$	" 12 Saturated brine	60	226.4

Pressure of Water due to its Weight.—The pressure of still water in pounds per square inch against the sides of any pipe, channel, or vessel of any shape whatever is due solely to the "head," or height of the level surface of the water above the point at which the pressure is considered, and is equal to .43302 lb. per square inch for every foot of head, or 62.355 lbs. per square foot for every foot of head (at 62° F.).

The pressure per square inch is equal in all directions, downwards, upwards, or sideways, and is independent of the shape or size of the containing vessel.

Table 28.—Pressure in Pounds per Square Inch for Different Heads of Water.

Head, feet.	0	1	2	3	4
0		0.433	0.866	1.299	1.732
10	4.330	4.763	5.196	5.629	6.062
20	8.660	9.093	9.526	9.959	10.392
30	12.990	13.423	13.856	14.289	14.722
40	17.320	17.753	18.186	18.619	19.052
50	21.650	22.083	22.516	22.949	23.382
60	25.980	26.413	26.846	27.279	27.712
70	30.310	30.743	31.176	31.609	32.042
80	34.640	35.073	35.506	35.939	36.372
90	38.970	39.403	39.836	40.269	40.702

Head, feet.	5	6	7	8	9
0	2.165	2.598	3.031	3.464	3.897
10	6.495	6.928	7.361	7.794	8.227
20	10.825	11.258	11.691	12.124	12.557
30	15.155	15.588	16.021	16.454	16.887
40	19.485	19.918	20.351	20.784	21.217
50	23.815	24.248	24.681	25.114	25.547
60	28.145	28.578	29.011	29.444	29.877
70	32.475	32.908	33.341	33.774	34.207
80	36.805	37.238	37.671	38.104	38.537
90	41.135	41.568	42.001	42.436	42.867

At 62° F. 1 foot head = 0.433 lb. per square inch, .433 × 144 = 62.352 lbs. per cubic foot.

Table 29.—Head in Feet of Water, Corresponding to Pressures in Pounds per Square Inch.

Pressure.	0	1	2	3	4
0		2.309	4.619	6.928	9.238
10	23.0947	25.404	27.714	30.023	32.333
20	46.1894	48.499	50.808	53.118	55.427
30	69.2841	71.594	73.903	76.213	78.522
40	92.3788	94.688	96.998	99.307	101.62
50	115.4735	117.78	120.09	122.40	124.71
60	138.5682	140.88	143.19	145.50	147.81
70	161.6629	163.97	166.28	168.59	170.90
80	184.7576	187.07	189.38	191.69	194.00
90	207.8523	210.16	212.47	214.78	217.09

Pressure.	5	6	7	8	9
0	11.547	13.857	16.166	18.476	20.785
10	34.642	36.952	39.261	41.570	43.880
20	57.737	60.046	62.356	64.665	66.975
30	80.831	83.141	85.450	87.760	90.069
40	103.93	106.24	108.55	110.85	113.16
50	127.02	129.33	131.64	133.95	136.26
60	150.12	152.42	154.73	157.04	159.35
70	173.21	175.52	177.83	180.14	182.45
80	196.31	198.61	200.92	203.23	205.54
90	219.40	221.71	224.02	226.33	228.64

1 lb. per square inch = 2.30947 feet head, 1 atmosphere = 14.7 lbs. per sq. inch = 33.94 ft. head.

Compressibility of Water.

Water is slightly compressible, its compressibility under a pressure of one atmosphere (14.7 lbs. per sq. in.) being from .000040 to .000051, decreasing with increase of temperature.

The incompressibility of water is so great that at the depth of a mile a cubic foot will only weigh about a half pound more than at the surface.

Table 30.—Specific Heat of Water.

0° C.	10°	20°	30°	40°	50°	60°	70°	80°	90°	100°
1.000	0.9943	0.9893	0.9872	0.9934	0.9995	1.0057	1.0120	1.0182	1.0244	1.0306

USEFUL FIGURES ON WATER.

One imperial gallon = 277.274 cubic inches.
" " = .16 cubic foot.
" " = 10.00 lbs.
" " = 4.537 litres.

One U. S. gallon = 231 cubic inches.
" " = .133 cubic foot.
" " = 8.33 lbs.
" " = .83 imperial gallon.
" " = 3.8 litres.

One cubic inch of water = .03607 lb.
" " " = .003607 imperial gallon.

One cubic foot of water = 6.23 imperial gallons.
" " " = 28.375 litres.
" " " = .0283 cubic metre.
" " " = 62.35 lbs.

One lb. of water = 27.72 cubic inches.
" " = .10 imperial gallon.
" " = .4537 kilo.

One cwt. of water = 11.2 imperial gallons.
" " = 1.8 cubic feet.

One ton of water = 35.9 cubic feet.
" " = 224 imperial gallons.
" " = 1000 litres (approximately).
" " = 1 cubic metre (approximately).

One litre of water = .22 imperial gallon.
" " = 61 cubic inches.
" " = .0353 cubic foot.

One cubic metre of water = 220 imperial gallons.
" " " = 1.308 cubic yards.
" " " = 61028 cubic inches.
" " " = 35.31 cubic feet.
" " " = 1000 kilos.
" " " = 1 ton (approximately).
" " " = 1000 litres.

One kilo. of water = 2.204 lbs.
One vedro of water = 2.7 imperial gallons.
One eimer of water = 2.7 imperial gallons.
One pood of water = 3.6 imperial gallons.
One Russian fathom = 7 feet.
One atmosphere = 1.054 kilos. per square inch.

A column of water 1 foot high = .434 lb. pressure per square inch.
A column of water 1 metre high = 1.43 lbs. pressure per square inch.
A pressure of 1 lb. per square inch = 2.31 feet of water in height.

BUOYANCY.

When a body is immersed in a liquid it is buoyed up by a force equal to the weight of the volume of liquid displaced by the body regardless of whether the body floats or sinks.

The weight of a floating body equals the weight of the liquid it displaces.

BOILING POINT.

Water boils at 212°F. (100°Cent.) at mean atmospheric pressure 14.7 lbs. per sq. in. at sea level. If water is entirely freed of air, which may be done by freezing, or boiling, the cohesion of its atoms is greatly increased, and its temperature may be raised 50° above the ordinary boiling point before it begins to boil.

Prof. Faraday found by experiment that when such air-freed water did boil, the rupture of the liquid resembled an explosion.

The boiling point of water will also be raised considerably above normal, if it is surrounded by a film of oil.

These facts have been utilized as a theoretical explanation of the cause of boiler explosions.

FUELS AND COMBUSTION.

The combustible and non-combustible ingredients of all fuels are,

COMBUSTIBLE.

1st. Fixed or free carbons.

2nd. Hydro-carbons, viz: Oleofiant gas, pitch, tar, etc.

NON-COMBUSTIBLE.

1st. Oxygen or hydrogen either in the form of water or in combination with the other constituents in the proportions that form water.

2nd. Nitrogen. Simply inert.

3rd. Sulphuret of Iron, only tending to cause spontaneous combustion.

4th. Other inert mineral compounds of various kinds, going to form the ash and clinker.

Heat Developed by Fuels.—The heat unit or British thermal unit (B. T. U.) is the quantity of heat required to raise the temperature of one pound of water one degree, or from 39° to 40° F., and the amount of mechanical work required to produce a unit of heat is 778 foot pounds. Therefore the mechanical equivalent of heat is the energy required to raise 778 lbs. one foot high, or 77.8 lbs. 10 ft. high, or 1 lb. 778 feet high. Or again, suppose a one-pound weight falls through a space of 778 ft. or a weight of 778 lbs. falls one foot, enough mechanical energy would thus be developed to raise a pound of water one degree in temperature, provided all the energy so developed could be utilized in churning or stirring the water, as in Joule's machine. Hence the mechanical equivalent of heat is 778 foot pounds.

Calorific Intensity.—The calorific intensity (C. I.) or pyrometric heating power is different from the calorific power. It is usually defined as the temperature which would be attained if combustion took place in exactly the right amount of oxygen, and all the heat were transferred to the products of combustion, and there were no loss. These conditions cannot be attained in practice, but the calculations are sometimes of importance for comparing the heating power of various fuels.

For instance, carbon, the C. P. is 14,500. These units of heat from 1 lb. of carbon have to heat 3.67 lb. of carbon dioxide which has a specific heat of .2163.

$$\text{The C. I.} = \frac{14{,}500}{3.67 \times .2163} = \frac{14{,}500}{.7938} = 18266°,$$ so that the temperature of the products of combustion would be $18{,}266 + 32 = 18{,}298°$.

For hydrogen the case is a little more complex, since the water has to be heated to 212° F., and then converted into steam and the steam heated further.

$$\text{C.I.}=\frac{61{,}500-[966+(1-.4805)\ 180]\ 9}{9\times.4805}=\frac{51{,}182}{4.3245}=11{,}835$$

rise of temperature.

For a fuel containing carbon, hydrogen, oxygen, and water, the formula is

$$\text{C. I.}=\frac{c\times14{,}500+[(h-\frac{1}{8}o)61{,}500-[966+(1-.4805)100\times(9h\times w)]}{(3.67c\times.2163+(9h+w)\times.4805)100}$$

Many other formulæ for calculating the calorific power of fuels, especially coals, have been suggested.

That of M. Cornut, modified to give the results in British units, is

$$\text{C. P.}=\frac{14{,}500\ c+20{,}390\ c_2+61{,}500\text{H}}{100}$$

Where c is the percentage of fixed carbon, c_2 the percentage of volatile carbon, and H the percentage of hydrogen.

Otto Gmelin formula, which does not require a complete analysis of the coal, is

$$\text{C. P.}=[100-(w+a)\ ]\ 80-c\times6\ w]\times1.8$$

where w is the percentage of water in the fuel, a the percentage of ash, and c a constant varying with the amount of water. The original result is given in calories, the multiplication by 1.8 converts it into British units.

The value of *c* is

For coal with water	less than	3	per cent	= 4
" "	between	3 and 4.5	"	= 6
" "	"	4.5 and 8.5	"	= 12
" "	"	8.5 and 12	"	= 10
" "	"	12 and 20	"	= 8
" "	"	20 and 28	"	= 6
" "	over	28	"	= 4

Calorific or Heating Power of Fuels.—The calorific or heating power of a fuel is the number of units of heat which will be evolved by the combustion of 1 lb. of fuel, the products of combustion escaping at a temperature considerably below 100° C. (212° F.).

For a fuel containing only carbon and hydrogen the calorific power is,

$$\text{C. P.} = \frac{c \times 14{,}500 + h \times 62{,}000}{100}$$

The calorific power of a fuel, the ingredients of which are carbon, hydrogen and oxygen, is,

$$\text{B. T. U.} = \frac{c \times 14{,}500 + (h - \frac{1}{8}\, o) \times 62{,}000}{100}$$

The heat developed by any fuel may be ascertained by calculation, when the chemical composition of the fuel is known. Table 31 gives the chemical compositions of various fuels.

One pound of pure carbon burning to carbonic acid develops 14,500 B. T. U. One pound of hydrogen develops 62,000 B. T. U. One pound of hydrogen requires 8 pounds of oxygen. Atmospheric air contains 20% of oxygen.

Table 31.—Chemical Composition of Combustibles.

Description.	Carbon	Hydrogen.	Oxyg'n	Ash.
Coke, average of 25 analyses	.935	....	....	.065
Charcoal, from wood, average	.925	....	....	.075
Oil of turpentine	.882	.118	....	....
Pine-wood oil	.867	.106	.027	....
Paraffin, or petroleum-oil	.855	.145	....	....
Petroleum, average	.840	.110	.050	....
Creosote, or tar-refuse	.825	.100	.075	....
Spermaceti	.817	.123	.060	....
Beeswax	.815	.139	.046	....
Coal, average	.804	.052	.079	.041
Schist oil	.800	.117	.083	....
Resin	.791	.102	.107	....
Asphalte	.790	.095	.087	.028
Tallow	.788	.118	.094	....
Sperm oil	.787	.110	.103	....
Olive oil	.770	.134	.096	....
Lignite, bituminous.	.749	.075	.130	.046
Lignite, perfect	.690	.050	.202	.058
Lignite, imperfect	.602	.053	.290	.055
Peat, perfectly dry	.600	.060	.300	.040
Peat, dry, average	.570	.060	.320	.050
Peat, dry, medium quality	.460	.050	.437	.053
Alcohol	.520	.135	.345	....
Wood-pimps, or bundles of twigs, dry	.510	.060	.400	.030
Wood, perfectly dry, average	.500	.060	.418	.022
Wood-bark, dry	.477	.063	.432	.028
Wood, pine, dry	.445	.055	.485	.015
Wood, oak, dry	.430	.053	.496	.021
Sawdust, dry	.410	.043	.531	.016
Straw, barley, dry	.380	.054	.510	.056
Straw, wheat, dry	.360	.050	.540	.050
Straw, oats, dry	.348	.050	.557	.045

In computing the heat value of combustibles containing oxygen, as well as carbon and hydrogen, the following principles are to be observed:

First: When hydrogen and oxygen exist in the proper proportion to form water (which by weight is one part of hydrogen to 8 parts oxygen), these constituents have no effect on the total heat of combustion.

Second: When hydrogen exists in a greater proportion, only the surplus of hydrogen above that which is required by the oxygen is to be taken into account.

Referring to Table 32, coke contains carbon .935, hydrogen .000, oxygen .000; therefore its maximum calorific power equals 14,500 × .935 equals 13,557 B. T. U.

Average coal, composed of carbon .804, hydrogen .052, oxygen .079, will develop 14,500 × .804 plus 62,000 × (.052 minus $\frac{.079}{8}$ equals 14,274 B. T. U.

Heating Power of Sawdust.—Sawdust, dry, average, is composed of .410 carbon, and contains about 1 lb. of hydrogen for each 8 lbs. of oxygen. The hydrogen being thus neutralized by the oxygen, it develops no heat. Consequently the calorific power of sawdust equals 14,500 × .410 equals 5,945 B. T. U.

Heating Power of Petroleum Oil.—As petroleum oil contains no oxygen, its calorific power equals 14,500 × .885 plus 62,000 × .145 equals 21,822 B. T. U.

Classification of Fuels.—

1. Solid fuels.
   - a—Natural fuels, wood, peat, coal.
   - b—Prepared, charcoal, peat charcoal, coke.
2. Liquid fuels.
   - a—Natural, oils.
   - b—Prepared, distilled oils.

3. Gaseous fuels.
   a—Natural, gas.
   b—Prepared, coal gas, producer gas, water gas, oil gas.

Table 32.—Calorific Values and Flame Temperatures of Solid and Gaseous Fuels (P. Mahler).

Substance.	Origin and Formula.	Calorific Value.		Flame Temperature.
		Gross Water Condensed.	Net Water as Vapour	
		B. Th. U. per lb.	B. Th. U. per lb.	Deg. F.
Oak Wood	Lorraine	8442	7866	3389
Peat	Bohemia	10620	10062	3668
Lignite	Trifail (Styria)	11970	11466	3560
Flaming Coal	Blanzy	15030	14508	3582
Flaming Coal	Decazeville	14112	13554	3560
Coal, oxidised by weathering	Commentry	11484	11160	3560
Coal, Gas	Commentry	15138	14598	3542
Coal, Gas	Bethune	15606	15084	3614
Coal, Gas	Lens	15732	15210	3650
Coal, Coking	St. Etienne	15948	15444	3650
Coal, Smithy	Roche la Moliera	15948	15480	3686
Coal, Semi-bituminous	Anzin	15588	15174	3596
Coal, Anthracite	Commentry	15228	14922	3686
Coal, Anthracite	Kebao, Tonkin	15354	15066	3668
Coal, Anthracite	Creusot	15642	15264	3650
Coal, Anthracite	Pennsylvania	14879	14652	3632
Alcohol, ethylic & methylic, or woodspirit	$C_2 H_5 (HO)$	12697	—	3092
	$C H_3 (HO)$	9553		
Alcohol, amylic	$C_5 H_{11} (HO)$	16232	—	3362
Petroleum, crude	American	—	18720	3632
Petroleum, spirit	American	—	18486	3488
Petroleum, refined	American	—	18594	3020
Hydrogen	H_2	61524	51804	3560
Carbon Monoxide	CO	4396	—	*3812
Methane	CH_4	24021	21591	*3362
Acetylene	$C_2 H_2$	21501	20752	*4262
Illuminating Gas	Average	—	—	*3542
Water Gas, Industrial	H_2 & CO	—	—	*3632

*Flame temperature of gases at constant pressure.

Table 33.—Heating and Evaporating Power of American Coals.

Coal From	Heat Units (B. T. U.) per lb.	Theoretical evaporation in lbs. of water from and at 212° F.	Heat Units (B. T. U.) per lb.	Theoretical evaporation in lbs. of water from and at 212° F.
	Of average poor quality.		Of average best quality.	
Arkansas	11757	12.16	12537	12.97
Illinois	9450	9.78	11700	12.12
Indiana, (Block)			10407	10.77
Indian Territory	11088	11.47	12800	13.25
Iowa	8702	9.01	11370	11.77
Kentucky			13345	13.81
Maryland			13700	14.18
Missouri			12656	13.82
New Mexico	9414	9.75	11756	12.17
Ohio, (Hocking Valley)			13309	13.78
Ohio				
Penn., Clearfield	11600	12.01	14000	14.49
" Pittsburgh			13104	13.46
" " (Gas)			13035	13.49
" " (Slack)			11739	12.15
" Wilkesbarre			13563	14.04
" Youghiogheny	12600	13.04	13480	13.95
Tennessee (Lump)	12215	12.65	12600	13.04
Tennessee			13167	13.63
Texas, (Ft. Worth)			11803	12.22
Virginia, (Pocohontas)	9450	9.78	13363	13.83
Washington			12316	12.75
West Virginia, (New River)			13374	13.84
" " (Best)	12800	13.25	14200	14.17

Table 34.—Heating Values, etc., of some British Coals.

Name or Location.	Moisture.	Ash.	Sulphur.	British Thermal Units Dry Coal.
Tredegar Peas	0.82	7.27	1 30	14,694
Nixon's Navigation	1.62	3.22	0.79	15,097
Bwllfa	0.7	5.2	..	14,666
North's Navigation	0.52	8.80	1 30	14,364
Rhymney	5.70	15.50	..	12,802
Abercarn Black Vein	1.28	8.47	..	12,880
Wearmouth	1.78	4.65	1.98	15,141
Ryhope	2.16	2.95	..	14,364
South Hetton	1.82	2.60	..	14,244
Lambton	1.65	4.15	..	14,675
Seaton Delaval	7.84	13.85	..	12,161
Preston	2.70	9.30	..	13,596
Pemberton	6.70	4.55	1.02	14,121
Wigan	4.64	2.75	1.24	14,275
Pendleton	3.90	1.91	0.63	14,500
Tyldesley	6.06	4.09	0.65	13,588
Bickershaw	4.36	1.96	1.07	13,725
Haydock	3.30	14.1	..	11,232
Arley	3.66	4.71	..	14,552
Tinsley Park	4.13	4.10	..	14,172
Silkstone	1 52	1 09	..	13,730
Staveley	2.77	2.55	1.71	13,480
Shipley	4.25	9.21	..	13,074
Shirebrook	11.32	9.95	..	12,448
Grassmoor	4.28	6.15	..	13,531
Shireoaks	4.83	10.8	1.17	12,189
Ansley Hall	9.25	4.15	..	12,508
Earl Dudley	7.95	6.20	..	12,802
South Wingfield	5.39	3.95	..	13,789
Bestwood	8.85	8.70	..	12,443
Cannock Chase	11.20	8.65	..	12,616
Pinxton Rough	7.19	8.45	..	12,568
Exhall Nuts	9.94	7.50	..	12,424
Pooley Hall	7.89	16.00	..	11,072
Whitwick	11.39	2.55	..	13,693
Clifton Colliery	9.90	5.21	2.06	12,855
Bannockburn Main	1.75	2.75	1.89	15,096
Hamilton	9.99	0.98	1.53	14,975
Longrigg Navigation	1.72	4.08	0.81	13,172
Kiltonge, Airdrie	10.45	1.41	1.65	13,795
Lower Drumgray Airdrie	7.14	1.30	1.45	13,926

Coke.—Coke is prepared by the destructive distillation of coal in ovens or retorts. Three kinds of coke are made.

(1.) *Soft Coke or Smithy Char.*—This is black and porous, and kindles easily; it is soft and brittle and will not stand great pressure. It is made at a low temperature.

(2.) *Hard or Furnace Coke.*—Varies in appearance, according to the method of manufacture. That made in beehive ovens has a dark grey color and a metallic lustre, and breaks with a columnar fracture. That made in by-product recovery ovens is black and lustreless and breaks into rhomboidal fragments. Made by either method it is hard, dense and compact, and has a specific gravity of about .9, and contains about 90 per cent of carbon.

(3.) *Gas Coke.*—This is made as a by-product in the manufacture of coal gas. It is usually very inferior in quality, because the coals used are selected on account of their gas-yielding, not coke-making properties.

The following analyses will give an idea of the composition of different kinds of coke:

	1 (dry) Smithy char.	2 (dry) Hard coke.	3 Gas coke.
Volatile matter	7.8	.....	.....
Fixed carbon	86.25	90.31	87.05
Ash	5.95	8.45	7.50
Sulphur	.735	1.246	1.95
Moisture	.....	.....	3.50

Coke always contains water. Sulphur is always present, but the percentage is almost invariably less than in the coal from which the coke was made.

Table 35.—Analyses of Coke.

(From report of John R. Procter, Kentucky Geological Survey.)

Where Made.	Fixed Carbon.	Ash.	Sulphur.
Connellsville, Pa. (Average of 3 samples)	88.96	9.74	0.810
Chattanooga, Tenn. " 4 "	80.51	16.34	1.595
Birmingham, Ala. " 4 "	87.29	10.54	1.195
Pocahontas, Va. " 3 "	92.53	5.74	0.597
New River, W. Va. " 8 "	92.38	7.21	0.562
Big Stone Gap, Ky. " 7 "	93.23	5.69	0.749

A Bushel of Coal.—The weight of a bushel of coal in Indiana is 70 lbs., in Pennsylvania, 76 lbs., in Alabama, Colorado, Georgia, Illinois, Ohio, Tennessee and West Virginia it is 80 lbs.

A Bushel of Coke is almost uniformly 40 lbs. in weight.

Table 36.—Experiments in Coking, Connellsville Region.

(John Fulton, *Amer. Mfr.*, Feb. 10, 1893.)

No. of Test.	Time in Oven.	Coal Charged.	Ash made.	Fine Coke made.	Market Coke made.	Total Coke made.	Per cent. of Yield.				Per Cent Lost.
							Ash.	Fine Coke.	Market Coke.	Total Coke.	
	hr.	lb.	lb.	lb.	lb.	lb.					
1	67	12,420	99	385	7,518	7,903	.80	3.10	60.53	63.63	35.57
2	68	11,090	90	359	6,580	6,939	.81	3.24	59.33	62.57	36.62
3	45	9,120	77	272	5,418	5,690	.84	2.98	59.41	62.39	36.77
4	45	9,020	74	349	5,334	5,683	.82	3.87	59.13	63.00	36.18
		41,650	340	1365	24,850	26,215	.82	3.28	59.66	62.94	36.24

Wood as Fuel.—Perfectly dry wood contains about 50% of carbon, the remaining 50% consisting almost entirely of oxygen and hydrogen in the proportions that form water.

The coneriferous family contains a small quantity of turpentine, which is a hydrocarbon. The heat value of wood as compared with coal may be safely assumed to be in the ratio of 2¼ to 1; that is, 2¼ lbs. of wood are equal in calorific power to 1 lb. of average quality soft coal.

Wood newly felled contains on an average 40% of water, and from 1% to 5% of ash.

Table 37.—Showing Proportion of Water Expelled from Wood at Gradually Increasing Temperatures. (M. Violette.)

Temperature.	Water Expelled from 100 Parts of Wood.			
	Oak.	Ash.	Elm.	Walnut.
257° Fahr.	15.26	14.78	15.32	15.55
302° Fahr.	17.93	16.19	17.02	17.43
347° Fahr.	32.13	21.22	36.94?	21.00
392° Fahr.	35.80	27.51	33.38	41.77?
437° Fahr.	44.31	33.38	40.56	36.56

Table 38.—Composition of Wood.
(Analysis of Woods, by M. Eugene Chevandier.)

Woods.	Composition.				
	Carbon.	Hydrogen.	Oxygen.	Nitrogen.	Ash.
	Per Cent.	Per Cent.	Per Cent.	Per Cent.	Per Cent.
Beech	49.36	6.01	42.69	0.91	1.06
Oak	49.64	5.92	41.16	1.29	1.97
Birch	50.20	6.20	41.62	1.15	0.81
Poplar	49.37	6.21	41.60	0.96	1.86
Willow	49.96	5.96	39.56	0.96	3 87
Average	49.70	6.06	41.30	1.05	1.80

The wood operated upon had been kept in store during two years. When wood which has been strongly dried by means of artificial heat is left exposed to the atmosphere, it reabsorbs about as much water as it contains in its air-dried state.

A cord of wood = 4 × 4 × 8 = 128 cu. ft. About 56% solid wood and 44% interstitial spaces. (Marcus Bull, Philadelphia, 1829. J. C. I. W., vol. i., page 293.)

B. E. Fernow gives the per cent of solid wood in a cord as determined officially in Prussia (J. C. I. W., vol. iii. p. 20):

Timber cords, 74.07% = 80 cu. ft. per cord.

Firewood cords (over 6″ diam.), 69.44% = 75 cu. ft. per cord.

"Billet" cords (over 3″ diam.), 55.55% = 60 cu. ft. per cord.

"Brush" woods less than 3″ diam., 18.52%; Roots, 37.00%.

Table 39.—Composition of Charcoal Produced at Various Temperatures. (By M. Violette.)

	Temperature of Carbonization.		Composition of the Solid Product.				
			Carbon.	Hydrogen.	Oxygen.	Nitrogen and Loss.	Ash.
	Cent.	Fahr.	Per Cent.	Per Cent.	Per Cent.	Per Cent.	Per Cent.
1	150°	302°	47.51	6.12	46.29	0.08	47.51
2	200	392	51.82	3.99	43.98	0.23	39.88
3	250	482	65.59	4.81	28.97	0.63	32.98
4	300	592	73.24	4.25	21.96	0.57	24.61
5	350	662	76.64	4.14	18.44	0.61	22.42
6	432	810	81.64	4.96	15.24	1.61	15.40
7	1023	1873	81.97	2.30	14.15	1.60	15 30

The wood experimented on was that of black alder, or alder buckthorn, which furnishes a charcoal suitable for gunpowder. It was previously dried at 150 deg. C. = 302 deg. F.

CHARCOAL.

Charcoal is made by evaporating the volatile constituents of wood and peat, either by a partial combustion of a conical heap of the material to be charred, covered with a layer of earth, or by the combustion of a separate portion of fuel in a furnace, in which are placed retorts containing the material to be charged.

According to Peclet, 100 parts by weight of wood when charred in a heap yield from 17 to 22 parts by weight of charcoal, and when charred in a retort from 28 to 30 parts.

This has reference to the ordinary condition of the wood used in charcoal-making, in which 25 parts in 100 consist of moisture. Of the remaining 75 parts the carbon amounts to one-half, or 37½% of the gross weight of the wood. Hence it appears that on an average nearly half of the carbon in the wood is lost during the partial combustion in a heap, and about one-quarter during the distillation in a retort.

Yield of Charcoal from a Cord of Wood.—From 45 to 50 bushels to the cord in the kiln, and from 30 to 35 in the meiler. Prof. Egleston in Trans. A. I. M. E., viii. 395, says the yield from kilns in the Lake Champlain region is often from 50 to 60 bushels for hard wood and 50 for soft wood; the average is about 50 bushels.

Various Solid Fuels.—Dust, when mixed with air, burns so rapidly as in some cases to cause explosions, as for instance, in confined passages in flour mills.

The explosive violence of the combustion of dust is due to the instantaneous heating and consequent expansion of the air.

Coal dust in mines promotes and extends explosions and will readily operate as a fiercely burning and explo-

sive agent through the medium of a very small proportion of fire-damp.

Pulverized fuel in the form of a powder may be burned without smoke, and with high economy.

The fuel, instead of being introduced into the fire-box in the ordinary manner, is first reduced to a powder by pulverizers of any construction. In the place of the ordinary boiler fire-box there is a combustion chamber in the form of a closed furnace lined with fire-brick, and provided with an air injector. The nozzle throws a constant stream of fuel into the chamber, scattering it throughout the whole space of the fire box. When this powder is once ignited, and it is very readily done by first raising the lining to a high temperature by an open fire, the combustion continues in an intense and regular manner under the action of the current of air which carries it in. (*Mfrs. Record,* April, 1893.)

Peat or Turf, as usually dried in the air, contains from 25% to 30% of water, which must be allowed for in estimating its heat of combustion. This water having been evaporated, the analysis of M. Regnault gives, in 100 parts of perfectly dry peat of the best quality: C 58%, H 6%, O 31%, Ash 5%.

In some examples of peat the quantity of ash is greater, amounting to 7% and sometimes to 11%.

The specific gravity of peat in its ordinary state is about 0.4 or 0.5. It can be compressed by machinery to a much greater density. (Rankine.)

Clark (Steam-engine, i. 61) gives as the average composition of dried Irish peat: C 59%, H 6%, O 30%, N 1.25%, Ash 4%.

GASEOUS FUELS.

Gaseous fuels are used with marked success in many localities. The most valuable gaseous fuel is the natural gas of Pennsylvania and Ohio, the calorific power being about 1,100 B. T. U. per cu. ft., or 10,000 calories per cu. metre. Compared with coal 57.2 lbs. of coal, or 63 lbs. of coke are equal in calorific power to 1,000 cu. ft. of natural gas.

ILLUMINATING-GAS.

Coal-gas is made by distilling bituminous coal in retorts. The retort is usually a long horizontal semi-cylindrical or D shaped chamber, holding from 160 to 300 lbs. of coal. The retorts are set in "benches" of from 3 to 9, heated by one fire, which is generally of coke. The vapors distilled from the coal are converted into a fixed gas by passing through the retort, which is heated almost to whiteness.

The gas passes out of the retort through an "ascension-pipe" into a long horizontal pipe called the hydraulic main, where it deposits a portion of the tar it contains; thence it goes into a condenser, a series of iron tubes surrounded by cold water, where it is freed from condensable vapors, as ammonia-water, then into a washer, where it is exposed to jets of water, and into a scrubber, a large chamber partially filled with trays made of wood or iron, containing coke, fragments of brick or paving-stones, which are wet with a spray of water. By the washer and scrubber the gas is freed from the last portion of tar and ammonia and from some of the sulphur compounds. The gas is then finally purified from sulphur compounds by passing it through lime or oxide of

iron. The gas is drawn from the hydraulic main and forced through the washer, scrubber, etc., by an exhauster or gas pump.

The kind of coal used is generally caking bituminous, but, as usually this coal is deficient in gases of high illuminating power, there is added to it a portion of cannel coal or other enricher.

Producer Gas.—This is made by passing air, or air and steam over red hot coke. When air is passed over red hot coke in excess, carbon monoxide is formed $C + O = CO$. Such a gas may be called simple producer gas. It contains

Carbon monoxide	34.7
Nitrogen	65.3
	100.0

and as both gases have the same specific gravity, the percentage is the same by weight as volume. 1,000 cubic feet of the gas weighs 78.2 lb., and 1 ton of carbon will give about 190,000 cubic feet of gas. Assuming coke to contain 90 per cent of carbon, 1 ton would yield 171,000 cubic feet. The calorific power of carbon monoxide burning to carbon dioxide being 4,368 B.Th.U., that of the gas will be $\frac{4368 \times 34.7}{100} = 1{,}515.69$.

Loss of Heat in Producer.—A considerable amount of heat is lost in the production of the gas. This is not lost in the sense that heat is made latent when water is boiled, but it is evolved in the producer where it is not wanted and thus is of no practical use.

1 lb. of carbon will give 2.33 lb. of carbon monoxide, and 6.71 lb. of simple producer gas.

1 lb. of carbon, burning to

carbon dioxide gives $14{,}500 \times 1 = 14{,}544$ B.Th.U.
6.71 lb. of producer gas
gives $6.71 \times 1515.7 = 10{,}170$

4,474

This 4,474 units, which is practically lost, amounts to about 31 per cent of the total heat available from the fuel.

Table 40.—Calorific Power of Gas Fuels.

AUTHORITY	GAS	B. T. U.
A. G. Glasgow, M. E.	Plain water gas.	327,268 per 1000 cu. ft.
A. C. Humphreys....	Plain water gas (theoretical).	323,003 per 1000 cu. ft.
F. E. Taylor, M. E...	Plain water gas.	8,335 per lb.
Dr. Greene	Plain water gas.	6,223 per lb.
Newbigging.....	Plain water gas.	290,000 per 1000 cu. ft.
	Plain water gas.	6,649 per lb.
Dr. Gideon Moore..	Carburetted water gas, 22 candle-power.	650,000 per 1000 cu. ft.
	Coal gas, 18 candle-power.	642,000 per 1000 cu. ft.
Newbigging.....	Coal gas. 17 candle-power.	673,224 per 1000 cu. ft.
	Coal gas, 17 candle-power.	21,696 per lb.
R. D. Wood & Co...	Coal gas.	735,000 per 1000 cu. ft.
	Water gas.	322,000 per 1000 cu. ft.
	Producer gas (anthracite).	137,000 per 1000 cu ft.
	Producer gas (bituminous).	156,000 per 1000 cu. ft.

Table 41.—Natural Gas in Ohio and Indiana.
(*Eng. and M. J.*, April 21, 1894.)

Description	OHIO			INDIANA			
	Fostoria	Findlay	St. Mary's	Muncie	Anderson	Kokomo	Marion
Hydrogen	1.89	1.64	1.94	2.35	1.86	1.42	1.20
Marsh-gas	92.84	93.35	93.85	92.67	93.07	94.16	93.57
Olefiant gas	.20	.35	.20	.25	.47	.30	.15
Carbon monoxide	.55	.41	.44	.45	.73	.55	.60
Carbon dioxide	.20	.25	.23	.25	.26	.29	.30
Oxygen	.35	.39	.35	.35	.42	.30	.55
Nitrogen	3.82	3.41	2.98	3.53	3.02	2.80	3.42
Hydrogen sulphide	.15	.20	.21	.15	.15	.18	.20

Compared with coal (approximately) 30,000 cu. ft. of gas = 1 ton of coal in heating power.

Table 42.—Producer-gas from One Ton of Coal.
(W. H. Blauvelt, Trans. A. I. M. E., xviii. 614.)

Analysis by Vol.	Per Cent.	Cubic Feet	Lbs.	Equal to—
CO	25.3	33,213.84	2,451.20	1,050.51 lbs. C+1,400.7 lbs. O.
H	9.2	12,077.76	63.56	63.56 lbs. H.
CH_4	3.1	4,069.68	174.66	174.66 lbs. CH_4.
C_2H_4	0.8	1,050.24	77.78	77.78 lbs. C_2H_4.
CO_2	3.4	4,463.52	519.02	141.54 lbs. C+377.44 lbs. O.
N (by difference)	58.2	76,404.96	5,659.63	7,350.17 lbs. Air.
	100.0	131,280.00	8,945.85	

Calculated upon this basis, the 131,280 ft. of gas from the ton of coal contained 20,311,162 B.T.U., or 155 B.T.U. per cubic ft., or 2,270 B.T.U. per lb.

The composition of the coal from which this gas was made was as follows: Water, 1.26%; volatile matter, 36.22%; fixed carbon, 57.98%; sulphur, 0.70%; ash, 3.78%. One ton contains 1159.6 lbs. carbon and 724.4 lbs. volatile combustible, the energy of which is 31,302,200 B.T.U. Hence, in the processes of gasification and purification there was a loss of 35.2% of the energy of the coal.

The composition of the hydrocarbons in a soft coal is uncertain and quite complex; but the ultimate analysis of the average coal shows that it approaches quite nearly to the composition of CH_4 (marsh-gas).

Table 43.—Average Analyses by Volume, and the Weight and Energy of the Four Types of Gases used for Heat and Light.

	Natural Gas.	Coal-gas.	Water-gas.	Producer-gas.	
				Anthra.	Bitu.
CO	0.50	6.0	45.0	27.0	27.0
H	2.18	46.0	45.0	12.0	12.0
CH_4	92.6	40.0	2.0	1.2	2.5
C_2H_4	0.31	4.0	.....	.....	0.4
CO_2	0.26	0.5	4.0	2.5	2.5
N	3.61	1.5	2.0	57 0	56.2
O	0.34	0.5	0.5	0.3	0.3
Vapor	.....	1.5	1.5	.....	.....
Pounds in 1000 cubic feet	%45.6	32.0	45.6	65.6	65.9
Heat units in 1000 cubic ft.	1,100,000	735,000	322,000	137,455	156,917

Water Gas.—When steam is passed over red-hot coke a mixture of carbon monoxide and hydrogen is obtained. This is called water gas. The following is an analysis:—

	By volume.
Hydrogen	49.17
Marsh gas	.31
Carbon monoxide	43.75
Carbon dioxide	2.71
Nitrogen	4.06
	100.00

The calorific power is about 7,524 B.T.U. 1,000 cubic feet weigh 43 lbs., and therefore would evolve 323,832 B.T.U. As the action of steam on coke absorbs a very

large quantity of heat, the gas can only be made intermittently. The fuel used is coke, air is blown in till the temperature is high enough, the producer gas either being allowed to escape or used. Then steam is blown in, and water gas made till the temperature falls, when air is again sent in.

Blast Furnace Gas.—The blast furnace is a gas producer on a large scale, the gas differing only from producer gas in containing rather more carbon dioxide. When coal is the fuel used, it is usually washed to remove and recover the tar and ammonia.

EXAMPLES OF BLAST FURNACE GASES.

	Coke fed.	Coal fed.	Charcoal.
Carbon dioxide	11.39	8.57	5.9
Carbon monoxide	28.61	27.15	29.6
Nitrogen	57.06	54.29	63.4
Hydrogen	2.74	5.48	.1
Hydrocarbon	.20	4.29	—

Gas for Gas Engines.—Gas for use in gas engines must be free from tar. There are three methods of insuring this.

(1) The gas may be made from coke or charcoal, which yield no tar. This is the method used in the Dowson producer.

(2) The tar may be destroyed. To effect this, the gas from the freshly distilled coal is made to pass downwards through the hot coke, and is thus broken up into carbon, and permanent gases. The destruction is often not complete, the gas leaving at a very high temperature, and causing loss of heat.

(3) The gas can be washed, and the tar and ammonia removed. This is by far the best method, and, when suit-

able coals are used, about 25 lb. of ammonium sulphate and 40 gallons of tar can be obtained for each ton of coal.

FLOW OF GAS IN PIPES.

The rate of flow of gases of different densities, the diameter of pipes required, etc., are given in King's Treatise on Coal Gas, vol. ii. 374, as follows:

If d = diameter of pipe in inches.
Q = quantity of gas in cu. ft. per hour,
l = length of pipe in yards,
h = pressure in inches of water.
s = specific gravity of gas, air being 1.

$$d = \sqrt[5]{\frac{Q^2 sl}{(1350)^2 h}},$$

$$h = \frac{Q^2 sl}{(1350)^2 d^5},$$

$$Q = 1350 d^2 \sqrt{\frac{dh}{sl}} = 1350 \sqrt{\frac{d^5 h}{sl}}.$$

Molesworth gives $Q = 1000 \sqrt{\frac{d^5 h}{sl}}$.

J. P. Gill, *Am. Gas-light Jour.*, 1894, gives:

$$Q = 1291 \sqrt{\frac{d^5 h}{s(l+d)}}.$$

This formula is said to be based on experimental data, and to make allowance for obstructions by tar, water, and other bodies tending to check the flow of gas through the pipe.

A set of tables in Appleton's Cyc. Mech. for flow of gas in 2, 6, and 12 in. pipes is calculated on the supposition

that the quantity delivered varies as the square of the diameter instead of as $d^2 \times \sqrt{d}$, or $\sqrt{d^5}$.

These tables give a flow in large pipes much less than that calculated by the formulæ above given, as is shown by the following example. Length of pipe 100 yds., specific gravity of gas 0.42, pressure 1-in. water-column.

	2-in. Pipe.	6-in. Pipe.	12-in. Pipe.
$Q = 1350\sqrt{\frac{d^5 h}{sl}}$	1178	18,368	103,912
$Q = 1000\sqrt{\frac{d^5 h}{sl}}$	873	13,606	76,972
$Q = 1291\sqrt{\frac{d^5 h}{s(l+d)}}$ ...	1116	16,327	93,845
Table in App. Cyc...	1290	11,657	46,628

An experiment made by Mr. Clegg, in London, with a 4-in. pipe, 6 miles long, pressure 3 in. of water, specific gravity of gas .398, gave a discharge into the atmosphere of 852 cu. ft. per hour, after a correction of 33 cu. ft. was made for leakage.

Substituting this value, 852 cu. ft., for Q in the formula $Q = C\sqrt{d^5 h \div sl}$, we find C, the coefficient, $= 997$, which corresponds nearly with the formula given by Molesworth.

Combustion.—Combustion is the chemical combination of substances, principally carbon and hydrogen, with oxygen. The weight of air theoretically necessary for the complete combustion of one pound of fuel equals .117 times the per cent of carbon plus .35 times the per cent of free hydrogen contained in the substance to be consumed.

Thus assume a fuel to contain, carbon 70%, hydrogen 3%, then $70 \times .117 + 3 \times .35 = 9.24$ lbs. air required. This is theoretically the quantity of air required, but in practice the quantity should be increased largely, in some cases doubled.

With the barometer at 30 in. 13 cu. ft. of air at a temperature of 60 degrees F. weigh 1 lb. and 12 lbs. of air are required to combine with the constituents of 1 lb. of coal, for perfect combustion, but to allow for working conditions 24 lbs. is necessary.

The reason for this is that the carbonic acid resulting from the combustion is not a supporter of combustion but must be diluted with at least half as much free air in order that combustion may be maintained.

Perfect combustion occurs only when all the combustible is burned, the result being the production of the highest stage of oxidation.

The temperature of the products of combustion may be calculated from the known weight of the fuel, and the products of combustion, the calorific power of the former, and the specific heat of the latter.

Table 44.—Specific Heat of Products of Combustion at Constant Pressure. (Rankine.)

Carbonic-acid gas	0.217
Steam	0.475
Nitrogen, probably	0.245
Air	0.238
Ashes	0.200

The proportion in which the substances noted in Table 44 occur in the products of combustion being known, the mean specific heat of all may be determined; and the

total heat of combustion of one pound of fuel being divided by the product of this weight multiplied by the mean specific heat, the quotient is the probable temperature of the furnace gases.

TOTAL HEAT OF COMBUSTION OF FUELS.
(Rankine.)

The following table shows the total heat of combustion with oxygen, of 1 pound of each of the substances named in it, in British thermal units, and also in pounds of water evaporated from 212°. It also shows the weight of oxygen required to combine with each pound of the combustible and the weight of air necessary in order to supply that oxygen. The quantities of heat are given on the authority of MM. Favre and Silbermann.

Table 45.

Combustible.	Pounds of oxygen per pound of combustible.	Pounds of air (about)	Total British heat units.	Evaporative power from 212° Fahr.
				Lb.
Hydrogen gas.....	8	36	62032	64.20
Carbon, imperfectly burned, so as to make carbonic oxide	1⅓	6	4400	4.55
Carbon, perfectly burned, so as to make carbonic acid...	2⅔	12	14500	15.0
Olefiant gas, 1 pound.....	3 3/7	15 3/7	21344	22.1
Various liquid hydrocarbons, 1 pound.....	.....	.....	from 21700 to 19000	from 22½ to 20
Carbonic oxide, as much as is made by the imperfect combustion of 1 pound of carbon,—viz., 2⅓ pounds.....	1½	6	10000	10.45

VALVES AND VALVE GEAR.

In a certain "catechism" of the locomotive the following question and answer appear: "Q. What is a locomotive? A. A locomotive is a boiler and two or more engines mounted on wheels." This answer, while not very definite, is certainly "short and to the point."

Two types of locomotive engines are in use, viz., simple and compound.

A simple engine, whether stationary or locomotive, is an engine in which the steam is made to do work in but one cylinder, after which it is exhausted into the atmosphere, or, as is the case with many stationary engines, the exhaust steam passes into a condenser in which a vacuum is maintained, and the steam is there condensed.

A compound engine is an engine in which the steam is made to do work in two or more cylinders before it is allowed to escape into the atmosphere, or condenser. The expansive properties of the steam are thus utilized in a much higher degree than with the simple engine, and great economy in fuel is the result.

As an entire section is devoted to compound locomotives, the subject will not be enlarged upon at this stage, but the attention of the student will now be directed to a study of the valves, valve gear, etc., of a simple engine.

Figs 1 and 2 show respectively a front end view of the cylinder, valve chest and saddle plate castings, and a section through the same parts showing the steam and exhaust passages. These castings require to be the best grade of iron, neither too soft nor too hard.

Cylinders and valve seats are generally cast together with the bed plates or bed castings A A, Figs. 1 and 2. Sometimes the bed castings are made separate from the

cylinders and in one piece, and the cylinders are then bolted to it about at the line B C, Fig. 1. The usual practice, however, is to cast one-half of the bed casting with each cylinder and then bolt the two halves together at the line DD, this being the center line of the engine.

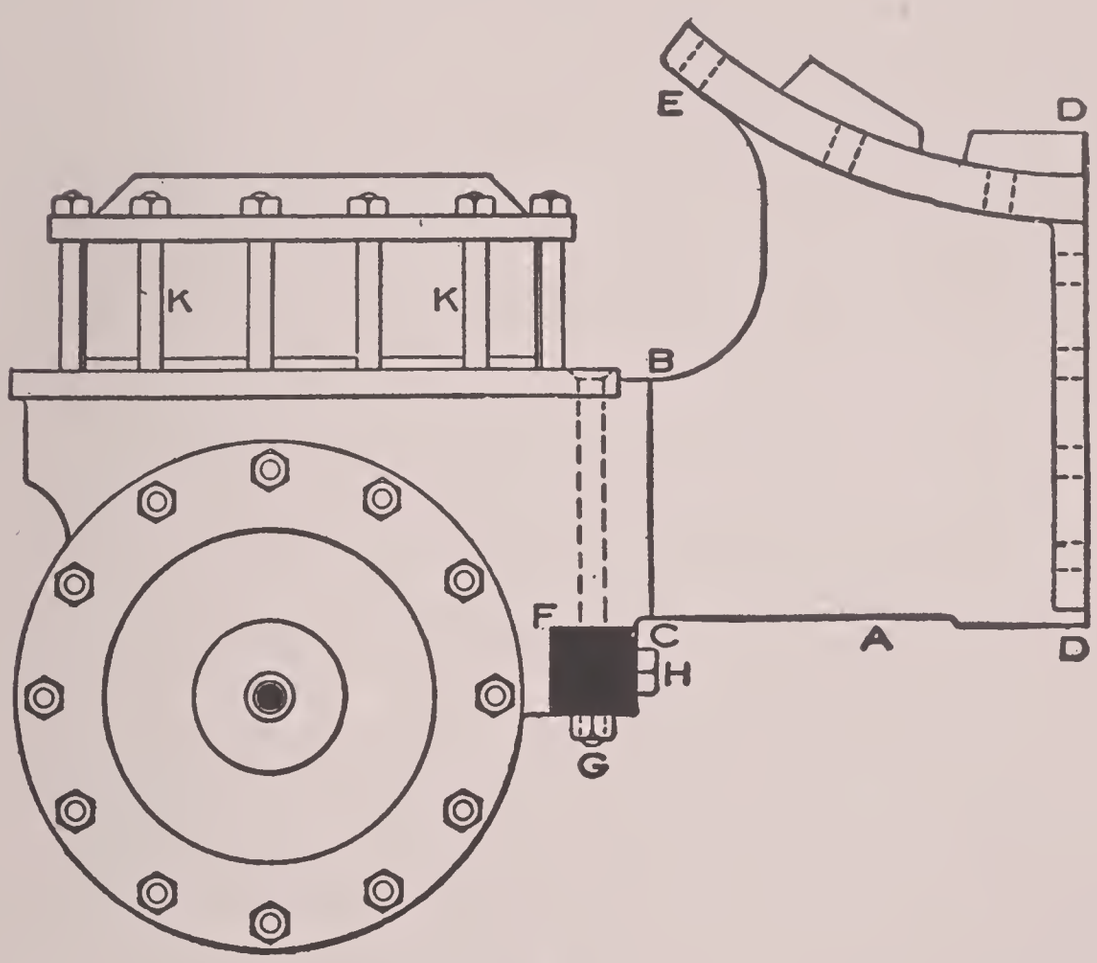


Fig. 1.

The bed castings are secured to the smoke-box by bolts through the flanges E E, and the cylinders are bolted to the frame F F by bolts G and H, Figs. 1 and 2. A structure is thus formed that is able to withstand the tremendous strains to which it will be subject.

By reference to Fig. 2 it will be seen that there are two passages in the bed casting leading to the cylinder. The

one L S is the live steam passage, and to this is connected the steam pipe and the other one *x x* is the exhaust passage leading from the cylinder to the smoke-box where the exhaust nozzle is attached.

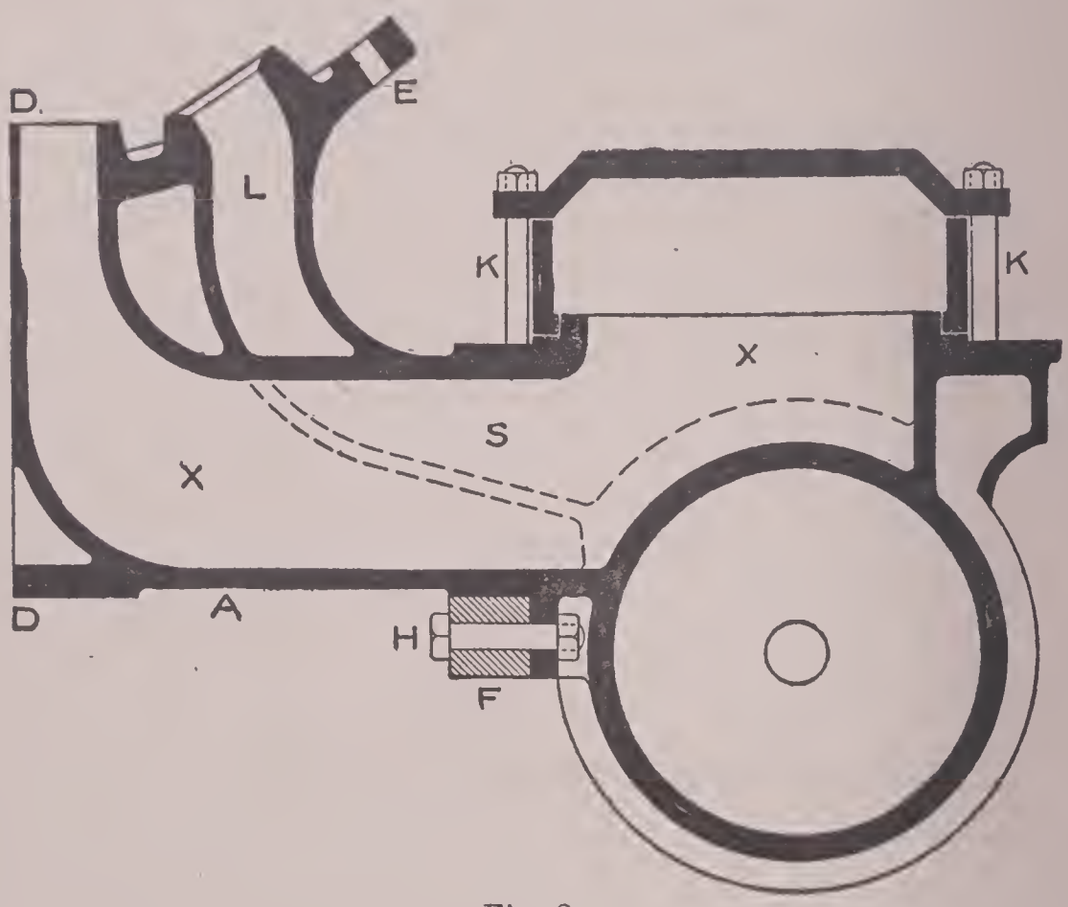


Fig. 2.

Fig. 3 gives a longitudinal sectional elevation of the cylinder, valve chest, valve, etc., showing the steam passages more clearly. Fig. 4 is a plan of the cylinder and guides and shows the valve seat with its ports, the steam chest cover and valve being removed. The steam passage A, on approaching the steam chest is divided into two branches, which terminate in openings B B at each end of the valve seat (see Figs. 3 and 4). This causes the steam to be delivered at both ends of the steam chest

and on top of the slide valve, which covers the ports P P and the exhaust port *x x* when in its central position (see Fig. 3).

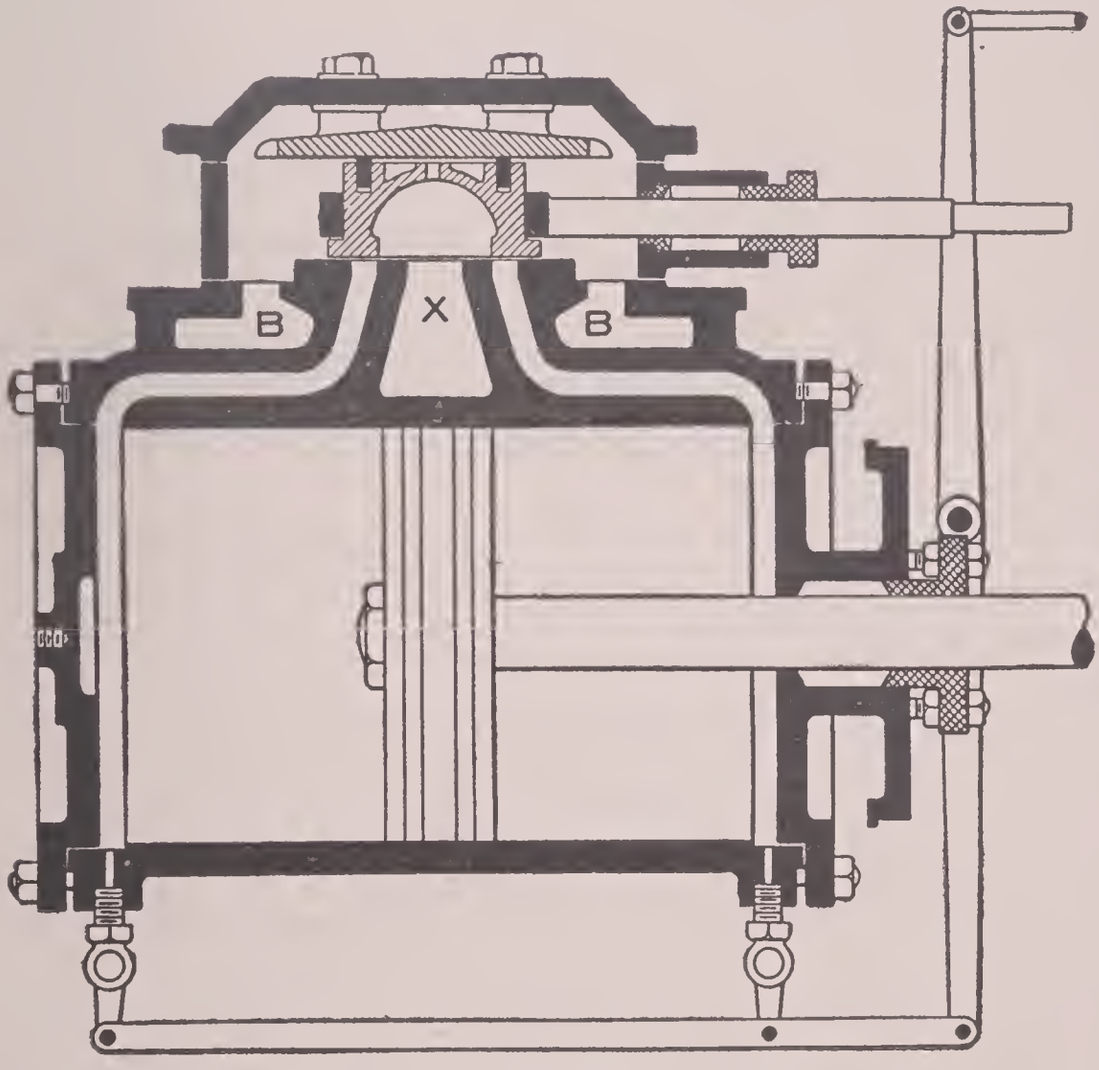


Fig. 3.

The steam chest is a square cast iron box open at top and bottom, and resting upon the top of the cylinder casting. The steam chest cover rests on top of this box and the whole is held down upon the cylinder by strong bolts K K, Fig. 1, forming a steam-tight joint, and within this valve chest the slide valve performs its important work.

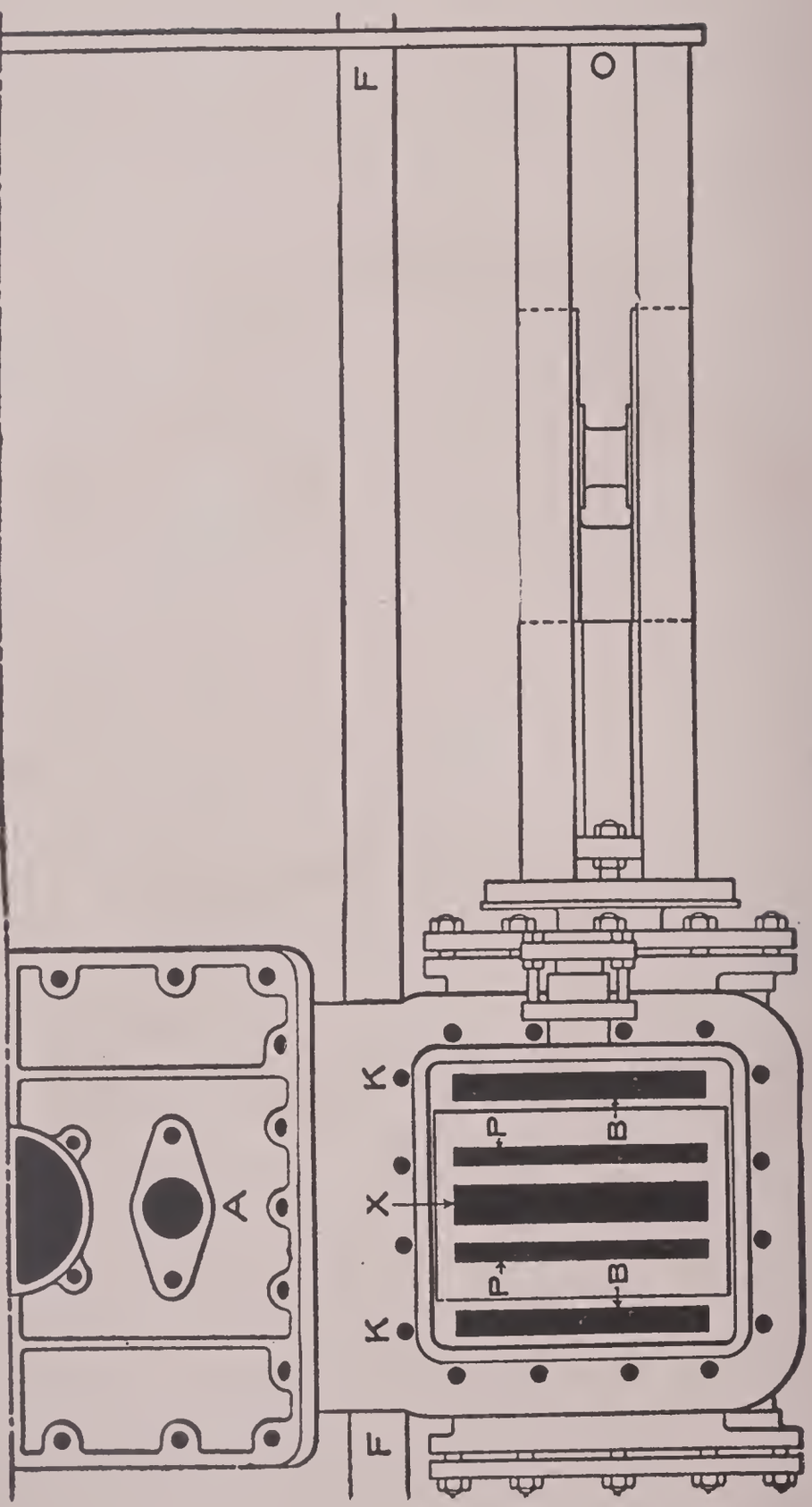


Fig. 4.

The invention of the D slide valve in its present form is the result of the investigations of Murdoch, who was an assistant to James Watt, the man who contributed more than all others towards the development of the steam en-

Fig. 5.

gine and its practical application. No young man who aspires to become a locomotive engineer should rest satisfied until he has obtained a thorough knowledge of the construction, operation, and adjustment of the slide valve.

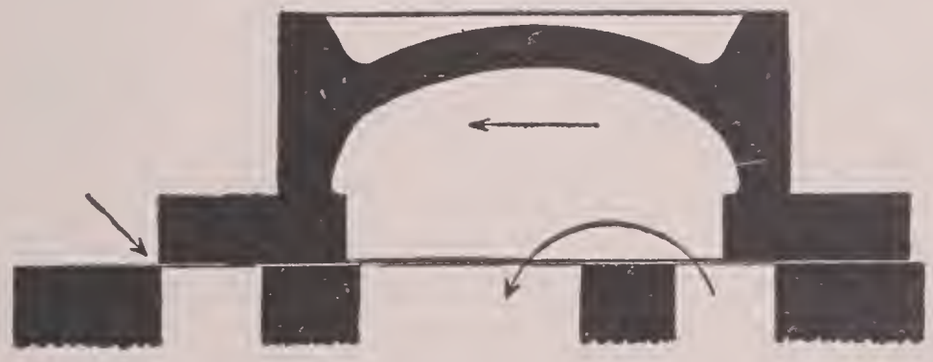


Fig. 6.

Many have been the efforts made to displace it with other types of valves, and while no doubt in stationary work other forms of valves may be better adapted to the conditions, yet the slide valve in some form or another still holds its own with the locomotive.

The functions that a slide valve must perform in order that the engine may do efficient work are five in number.

and they are as follows: First, it must admit steam into one end only of the cylinder at the same time. Second, it must cover the steam ports so as not to permit the passage of live steam through both steam ports at the same time. Third, it must allow the steam to escape from one end of the cylinder before it is admitted at the other end, so as to give the steam that is to be exhausted time to escape before the piston commences the return stroke. Fourth, it must not permit live steam to enter the exhaust port direct from the steam chest. Fifth, it must close each steam port on the steam side before it is opened on the exhaust side; this is for the purpose of utilizing the expansive force of the steam.

Fig. 7.

Figs. 5 to 10, inclusive, show the general construction of the D slide valve and illustrate the various positions assumed by it during one stroke. Fig. 5 shows the valve in its central position and explains the meaning of the word lap. Outside lap, often referred to as steam lap, is the distance that the edge of the valve overlaps the steam ports when it stands central or at mid-travel, and is that portion of the valve marked L and indicated by the distance between the lines O P.—Inside lap, frequently referred to as exhaust lap, is that portion of the valve that overlaps the two bridges of the valve seat

when the valve stands central, and is shown at A and A, Fig. 11.

Inside clearance, sometimes called exhaust lead, is the space between the inside edges of the exhaust arch of the valve and the bridges when the valve stands central. It means just the reverse of inside lap; that is, the distance between the inside edges of the exhaust arch is slightly greater than the distance between the inside edges of the steam ports, so that it does not entirely cover them when

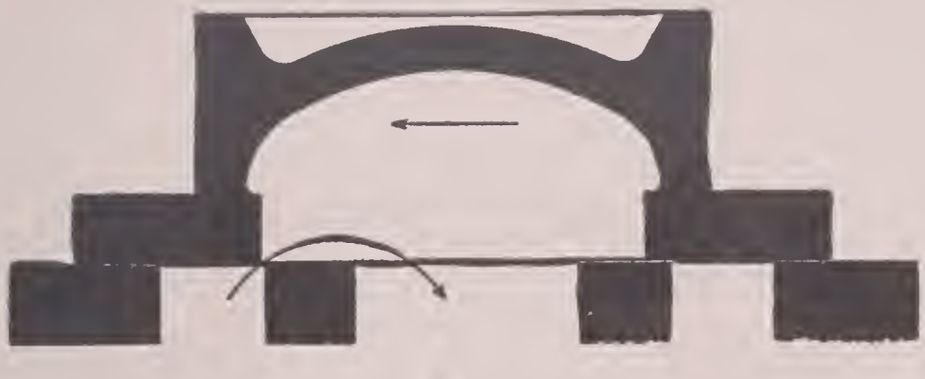


Fig. 8.

in its central position. Inside clearance is shown at B and B, Fig. 12. The purpose of inside lap is to delay the release of the steam and to hasten compression. The amount of inside lap is small, seldom exceeding ⅛ of an inch, and for fast passenger engines it is better to have none, as it causes the engine to be quicker. The purpose of inside clearance is exactly the opposite to that of inside lap. It hastens release and delays compression. It rarely exceeds 1-64th of an inch, and is only used on very fast running engines. Good judgment and great experience are required in order to determine the proper amount of inside clearance and upon what classes of locomotives it should be used.

In locomotives for ordinary service the valves have no inside clearance. Cut-off, Fig. 6, refers to the closing of the steam port by the valve, thereby cutting off the flow of live steam to the cylinder before the piston has completed its stroke. Compression refers to the early closure of the passage between the cylinder and the exhaust port. This point is reached when the inside or exhaust edge of the valve has closed the steam port, as shown in Fig. 7, wherein the valve is assumed to be traveling in the direction indicated by the arrow. A small portion of the steam is thus retained in the cylinder to be compressed by the advancing piston, which thus meets with a slight cushion at the end of its stroke, and all shock and jar is thus prevented. Release occurs when the exhaust edge of the valve opens the steam port and allows the steam that has completed its work in the cylinder to escape into the exhaust port, as shown in Fig. 8.

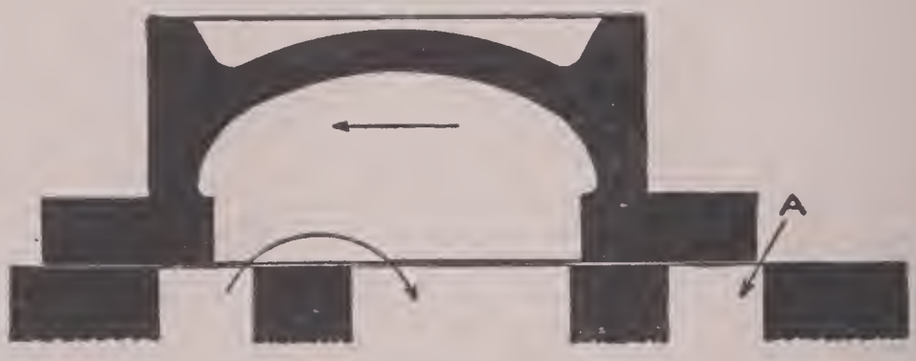


Fig. 9.

Lead, otherwise called steam lead, is the amount of opening given to the steam port by the valve for the admission of live steam to the cylinder when the piston is at the commencement of its return stroke. The lead is indicated by the letter A in Fig. 9.

Travel is the distance through which the valve travels, otherwise its stroke. Over travel is the distance the steam

edge of the valve travels after the steam port is wide open, indicated by distance between lines O and T, Fig. 10.

The objects aimed at in giving a valve outside or steam lap are: First, that the steam may be cut off before

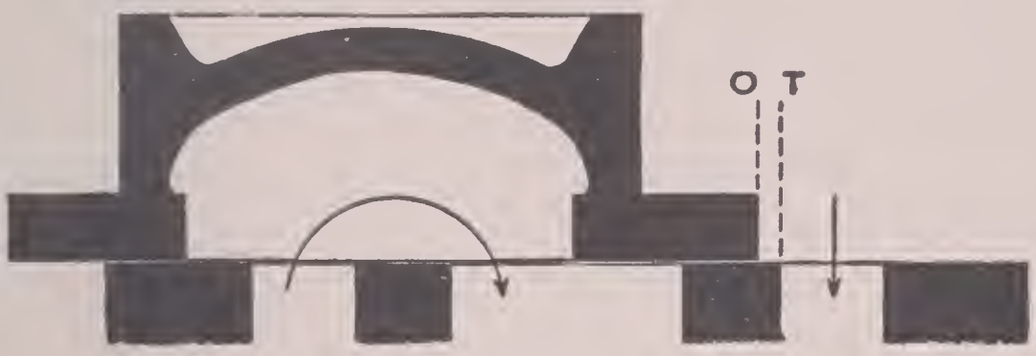


Fig. 10.

the piston reaches the end of its stroke, and the steam thus enclosed within the cylinder be made to do the work throughout the remainder of the stroke by reason of its expansive properties, and secondly, it causes the exhaust

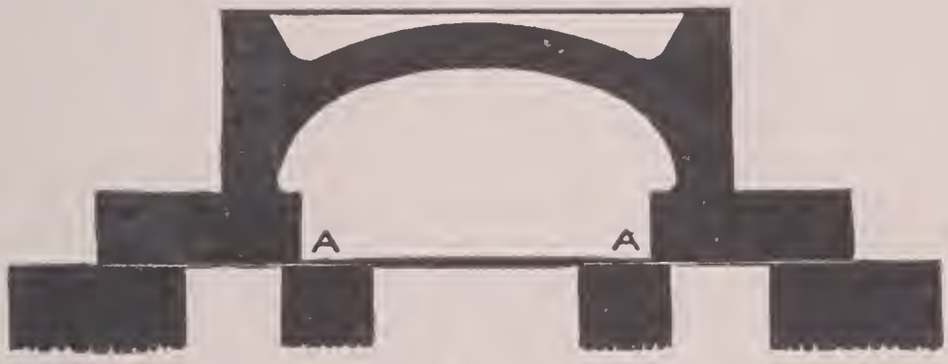


Fig. 11.

port at one end of the cylinder to be opened before the steam port at the other end is uncovered for the admission of steam. If a valve had no outside lap it would admit steam throughout the whole stroke, or in other words, "follow full stroke." Another bad effect of no lap would be a late exhaust, by which is meant that the exhaust would occur at one end of the cylinder at practically the

same moment that admission occurred at the other end. This would have a tendency to retard the motion of the piston.

The term clearance, as applied to a locomotive, means all of the space between the face of the valve and the piston when the latter is at the end of its stroke. Mechanical clearance means the distance between the cylinder head and the piston when at the end of its stroke.

The object of giving a valve lead is that the steam port may be opened slightly for the admission of live steam just before the piston reaches the end of its stroke, in order that there may be a cushion of steam to receive the

Fig. 12.

piston and reverse its motion at the end of the stroke, thus making the engine quicker in its action. Lead increases on a locomotive as the cut-off is made earlier or shorter, which is done by bringing the reverse lever nearer the center notch of the quadrant. The increased lead is caused by the radius of the link. This will be explained later on, towards the close of this section.

The term valve gear, as applied to a locomotive, includes eccentrics, rods, links, rockers, etc., by which the valves are given motion and by which their movements are regulated and controlled. As it is very necessary that a locomotive should be capable of being moved by steam either backward or forward, a reversible valve gear is re-

quired. Various devices have been invented for this purpose, but the shifting link motion has, after many years' trial, been found to be the most reliable and best and is to-day the standard in this country.

Fig. 13 shows a general view of the valve gear of one side of a locomotive. The center of the go-ahead eccentric is shown at A, and the center of the back-up eccentric is at B. The eccentric straps are shown connected to the eccentric rods, or blades as they are usually termed, and these in turn are attached to the link, the go-ahead eccentric being connected to the top end of the link and the back-up eccentric to the bottom end. The link saddle S is a plate spanning the center of the link and securely bolted to it. Upon the saddle a pin is formed, to which the lower end of the link hanger is connected, the top end of the hanger being attached to the shorter arm of the tumbling shaft, while the other arm of the tumbling shaft is connected to the reversing rod which extends back to the cab and is connected to the reverse lever.

The link and ends of the eccentric rods connected to it are thus supported and are also free to be moved either up or down by means of the reverse lever. The link block, upon which the link slides freely, is attached by a pin to the lower arm of the rocker shaft, and the valve rod is connected to the top arm of the rocker. The rocker shaft rotates in the rocker box, which is rigid, and the motion of the eccentric is thus indirectly imparted to the valve; that is, the motion of the eccentric is reversed by the rocker arm. This is termed indirect valve gear and is the standard type in this country, although there are some engines fitted with direct valve gear in which both arms of the rocker shaft extend upwards, or in the same direction, in which case the motion is not reversed.

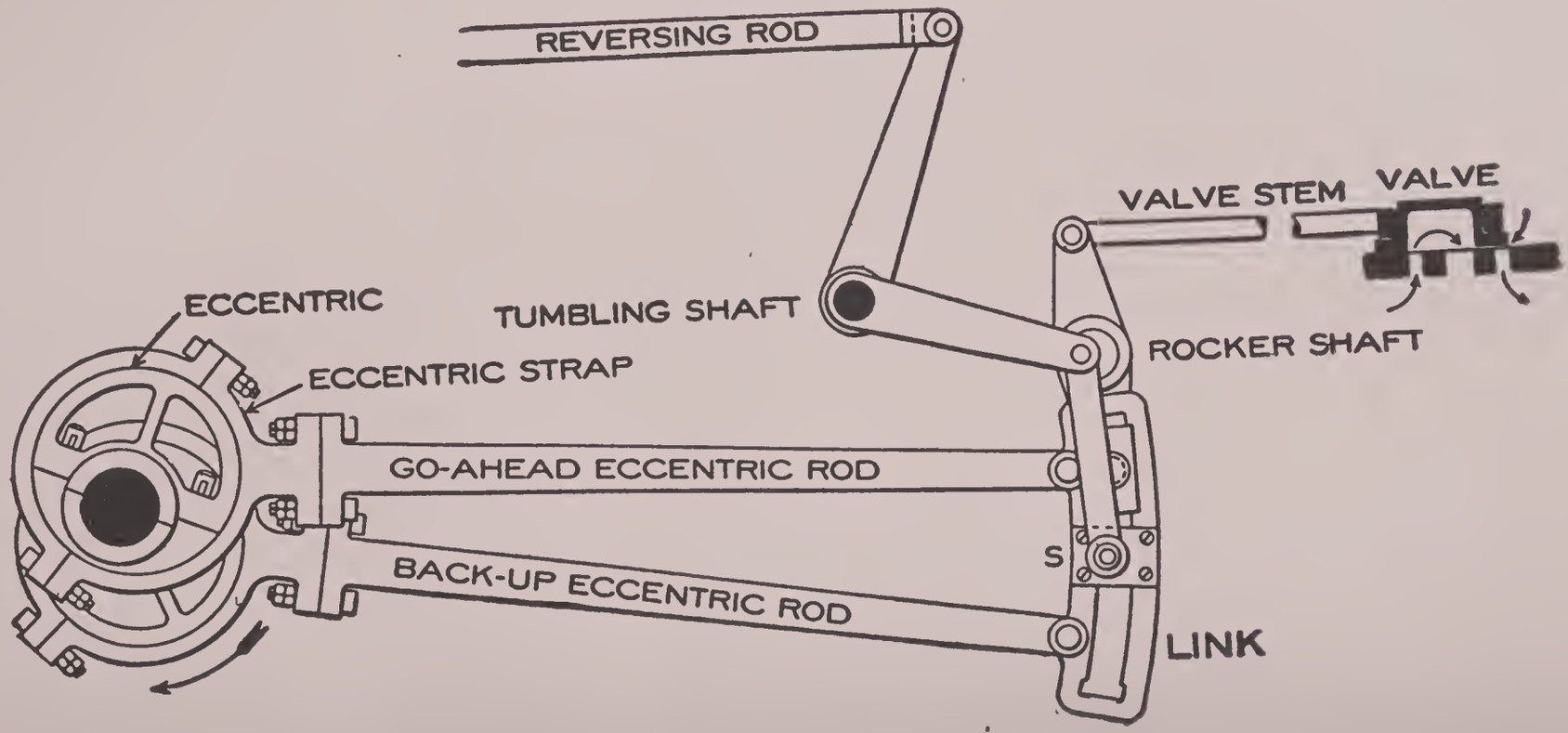

REVERSING ROD
VALVE STEM
VALVE
ECCENTRIC
TUMBLING SHAFT
ROCKER SHAFT
ECCENTRIC STRAP
GO-AHEAD ECCENTRIC ROD
S
BACK-UP ECCENTRIC ROD
LINK

Fig. 13.

The valve is connected to the valve stem usually by a yoke or frame that loosely embraces the top of the valve which is formed to receive it. This allows the valve to change its position vertically, with respect to the valve stem, as the face of the valve and the seat wear away. Sometimes the valve is secured by nuts that engage with a thread on the stem. In any case it is essential that the valve should have a small amount of freedom in its connection with the stem in order to guard against its becoming cocked or tilted on its seat, thus allowing the steam to blow past it.

The cut Fig. 13 shows the link in full forward gear; that is, the full throw of the go-ahead eccentric A affects the link block and rocker arm and through these the valve. By throwing the reverse lever back into the extreme back notch of the quadrant the link will be raised until the pin that connects the lower or backing eccentric blade to the link will be in line with the pin of the link block, which will then be affected by the full influence of the back-up eccentric B, and the engine will run backward.

It has been previously stated that the lead was increased by bringing the reverse lever nearer the center notch of the quadrant, or in common phraseology "hooking her back," in order to cause cut-off to take place earlier, and that the increase in lead was due to the radius of the link. The radius of the link is the distance, on a horizontal line, from the center of the main driving shaft, which carries the eccentrics, to the center of the rocker shaft. Ordinarily an increase in the lead is obtained by moving the eccentrics ahead on the shaft, but in this case the eccentric straps are moved back on the eccentrics by raising the link, as will readily be seen by a study of the diagram, Fig. 13.

The nearer the center of the link—that is, the center of the saddle pin—is brought to the center of the link block, the more are the eccentric straps moved back upon the eccentrics and the shorter will be the cut-off and the greater the lead.

With locomotives having very long eccentric blades there will not be such a marked increase in the lead as with those having very short blades, for the reason that the short rods will cause the straps to move farther around and back on the eccentrics by raising the link to which the ends of the rods are connected, whereas if the rods were longer their ends at the link could be raised considerably and still not materially affect the positions of the eccentric straps. This would indicate that in setting the valves of a locomotive, when it comes to adjustments for lead, attention should be paid to the radius of the link, which, as before stated, is the distance from the center of the main driving shaft to the center of the rocker shaft.

Authorities differ in regard to the amount of lead that locomotive valves should have at full gear. The older practice was to give an eighth of an inch, but of late years the tendency has been to cut it down to a sixteenth, or a thirty-second, and some authorities recommend even negative lead, which means no lead at full gear. They claim that too much lead is detrimental to an engine, causing more wear and tear to the valve gear, also that the pre-admission of steam is too great at mid-travel. With a 4-ft. radius the valves should be set line and line, (no lead) forward and back gear. With a 6-ft. radius one-sixteenth of an inch is required, and with a radius of 8 ft. the valves should have one-eighth of an inch positive lead forward and back gear. The travel of the valve is also reduced by hooking the reverse lever back from either full gear towards the center notch of the quadrant.

Questions.

1. What is a simple engine?
2. What is a compound engine?
3. Why is a compound engine more economical in fuel than a simple engine is?
4. Describe the saddle plate of a locomotive.
5. How is this casting secured to the smoke box?
6. What supports the cylinders?
7. How does the steam pass from the steam pipes in the smoke-box to the valve chests?
8. Describe the steam chest.
9. Who invented the D slide valve?
10. How many functions must a slide valve perform?
11. What is the first of these?
12. What is the second function?
13. Describe the third function.
14. What must the valve do in the fourth function?
15. What is the fifth function of the slide valve?
16. What is outside lap?
17. What is inside lap or exhaust lap?
18. What is inside clearance or exhaust lead?
19. What is the purpose of inside lap?
20. How much inside lap is usually given a locomotive slide valve?
21. What is the purpose of inside clearance?
22. What class of engines is it used on?
23. Do the valves of locomotives in ordinary service have or need inside clearance?
24. What is meant by cut-off, as applied to a slide valve?
25. What is compression?
26. When does compression begin?

27. What advantage is there in compression?

28. When does release occur?

29. What is steam lead?

30. What is meant by valve travel?

31. What is over travel?

32. What are the objects aimed at in giving a valve lead?

33. What would be the result if a valve had no lead?

34. Name another bad effect that would occur if a valve had no lead?

35. What is meant by clearance?

36. What is mechanical clearance?

37. What causes the lead on a locomotive valve to increase when the reverse lever is hooked back towards the center notch?

38. What does the term valve gear include, as applied to a locomotive?

39. What kind of a valve gear does a locomotive require?

40. Why are two eccentrics needed on each side of a locomotive?

41. What is the link saddle?

42. How is the link and the ends of the eccentric rods that are connected to it supported?

43. For what purpose is the link block?

44. What is an indirect valve gear?

45. What is a direct valve gear?

46. How is the slide valve usually connected to the valve stem?

47. What other method is sometimes used?

48. Why should a slide valve have a small amount of freedom in its connection?

49. What effect does it have upon the cut-off when the reverse lever is brought back towards the center notch?

50. What is the radius of the link?

51. Explain why it is that when the reverse lever is hooked back the lead increases.

52. How does this affect locomotives having long eccentric blades?

53. About how much lead is usually given an engine at full gear?

54. How should the valve be set when the radius of the link is 4 ft.?

55. What should the lead be with a 6-ft. radius?

56. How much lead should be given the valves when the radius is 8 ft.?

57. How is the travel of the valve affected by hooking the reverse lever back?

THE WALSCHAERT VALVE GEAR.

As previously stated the shifting link valve gear, or Stephenson link motion as it is sometimes termed, is at present, and has been the standard valve gear for locomotives in this country for many years, nevertheless a description of valve gear would not be complete, without including some, at least, of the various types of mechanism that have been invented and applied to the locomotive for the purpose of giving motion to the valves.

Probably the most prominent, and meritorious of these is the Walschaert valve gear, now largely used by some of the leading railroads.

GENERAL DESCRIPTION.

The Walschaert valve gear differs from the Stephenson link motion in that it requires for each cylinder but one eccentric or its equivalent, to insure the movement of the valve, and the proper distribution of the steam for both forward and backward motion. The elimination of the heavy eccentrics and their connections relieves the axle of an appreciable portion of its dead weight. It also makes it possible to place the gear outside the driving wheels in a more accessible position, where the parts can readily be oiled, inspected and repaired.

While it may not be possible to adjust the valve as readily with the Walschaert gear as with the Stephenson motion, for the reason that the parts and connections are not as susceptible to change, it is not as liable to become

disarranged, and if correctly designed and fitted up will give accurate results with less difficulty in application and greater economy in construction.

Equal cut-offs in both ends of the cylinder are more easily secured than with the Stephenson motion, and the play of the engine on its springs has practically no influence on the steam distribution.

In the Walschaert valve gear the operating eccentric is secured to the driving axle either directly or by a return crank from one of the crank pins.

The position of this eccentric or crank is such as to give the proper valve travel, the throw corresponding with the movement of the valve irrespective of its lap and lead; the angular advance of the eccentric being 0°.

The link is of any convenient form and is usually pivoted to a support on the engine frame or suspended from the guide bearer. The trunnion is rigid and there is no chance for twisting strains. The link is actuated by the eccentric rod which is commonly attached to its lower extremity. The sliding block in the link is secured to one end of the radius rod. The raising or lowering of this rod by means of the reversing shaft, shifts the block from one end of the link to the other above or below the pivoted connection; this reverses the movement of the valve with relation to that of the eccentric.

In many cases the gear can be so designed that the motion is transmitted from the eccentric to the valve stem in one vertical plane so that practically all of the pins can be put in double shear, and all tendency to twist the valve motion is avoided. In some cases (see Fig. 22) it is necessary to have the eccentric and the valve stem in different vertical planes. In such cases some form of rock shaft is necessarily employed to transfer the motion

from one plane to the other and give the required solidity to the gear. It is sometimes urged against the Walschaert motion that it requires more moving parts than the Stephenson. This is, however, offset by the better opportunities for solid construction and ready inspection and adjustment.

The end of the radius rod opposite the link is attached to a combining lever, the function of which is to give the required lap and lead to the valve. The lower end of this lever is connected to and travels with the crosshead, while to the upper end is secured both the valve rod and the radius rod, one being placed above the other. The point at which the radius rod is attached to the combining lever becomes a fulcrum. The relative movement of the two ends of the lever must be such that the full movement of the crosshead imparted to the lower end of the lever will give a movement of the upper end equivalent to twice the required lap plus the lead.

Under ordinary conditions with steam chest valves having outside admission, the connection or fulcrum between the radius rod and the combining lever is placed below the valve rod connection. With valves having inside admission, this fulcrum is usually placed above the valve rod connection as shown in Fig. 14.

The link should have a radius equal to the length of the radius rod. If this is so it will be seen that when the engine is on the dead center the link block can be moved from end to end of the link without altering the position of the valve with relation to the ports and the lead will be constant.

As any variation in the length or relative position of the link, the radius rod, and the combining lever or its connections, will necessarily change the resulting move-

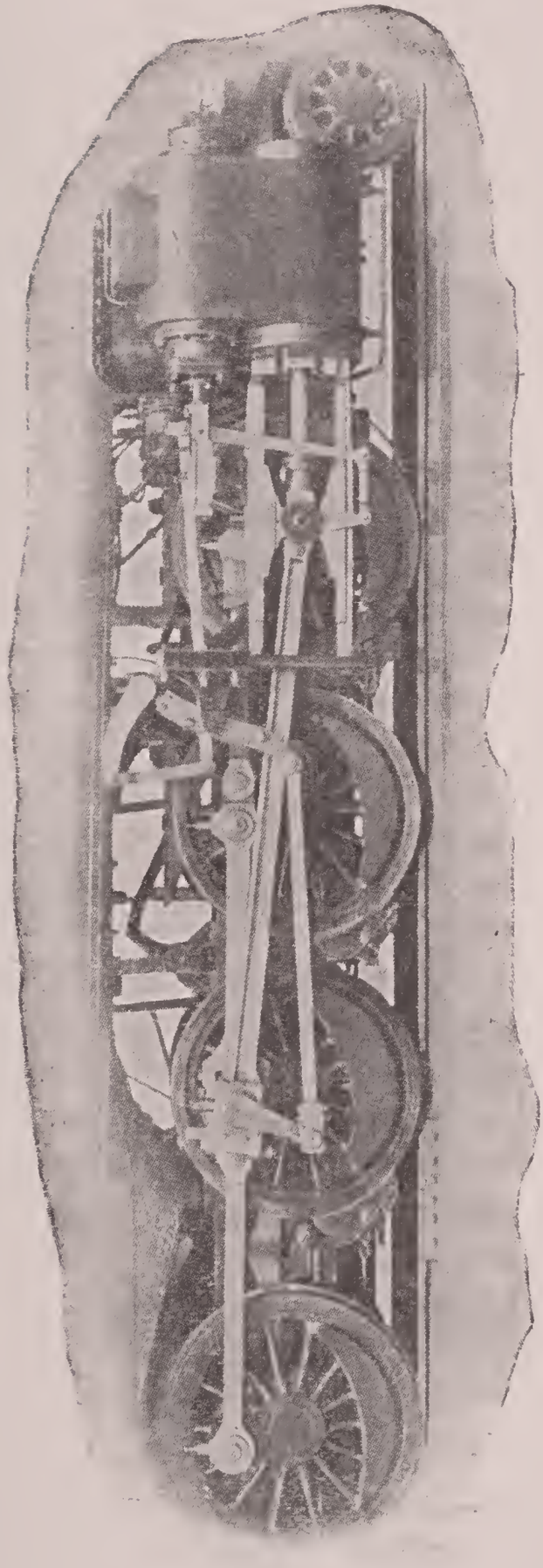


FIGURE 14. WALSCHAERT'S VALVE GEAR AS APPLIED TO CONSOLIDATION LOCOMOTIVE FOR THE PENNSYLVANIA RAILROAD.
Baldwin Locomotive Works.

ment of the valve; it is absolutely essential, first, that the motion shall be correctly designed and plotted, and second, that the detail parts shall be accurately constructed according to the diagram. With these two points assured the adjustment of the gear on the locomotive is quite simple. The dead center marks on the rim of the driving wheel and the port tram marks on the valve stem are found in the usual manner. After connecting the gear any slight variation which may occur between the forward and backward position of the valve can be adjusted by lengthening or shortening the eccentric rod.

The size and arrangement of parts in a modern locomotive make it difficult for an engineer to properly examine the eccentrics and link motion when the engine is on the road, and breakdowns are more frequent on this account. The conditions of service also tend to make it more and more difficult for the engine man to give the close inspection and care which is demanded in other branches of engineering service with high speed machines. Stationary engine practice may serve as a guide to those responsible for the successful operation of locomotives, which are even to a greater extent than stationary engines, high speed machines. With the Walschaert valve motion only a single eccentric or its equivalent is necessary for each valve. As usually constructed it is found more convenient to substitute a return crank, thus reducing the pin bearings to the smallest possible diameter, so that they may be readily lubricated, and, owing to the small amount of work they have to do, give satisfactory service and absolute freedom from heating.

So far as the distribution of steam in the cylinders is concerned, the constant lead, which is a feature of this

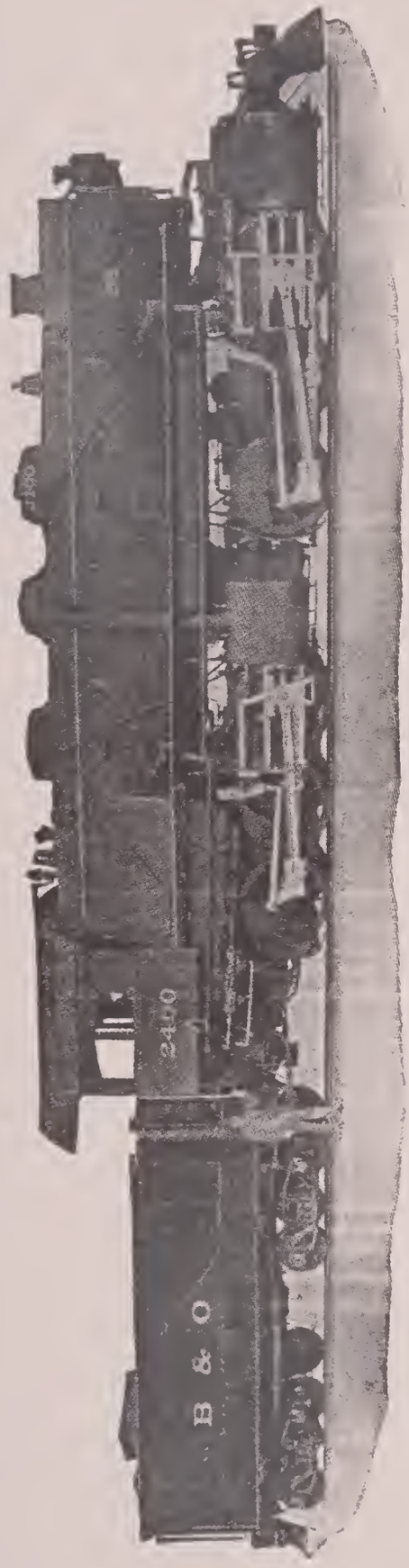


FIGURE 15. MALLET ARTICULATED COMPOUND FREIGHT LOCOMOTIVE WITH WALSCHAERT VALVE GEAR.
Built for Baltimore & Ohio Railroad. (American Locomotive Co.)

motion is not considered objectionable, and it has some distinct advantages. Under such conditions it is possible to determine the amount of lead the engine should have at the most economical point of cut-off. This point determined, and so designed, it cannot be altered by anyone in the shops or roundhouses. Another advantage is that it prevents valve setters from attempting to produce results by moving the eccentrics into improper relations one to another.

The constant lead of the Walschaert motion prevents the sealing of the cylinders by the piston valve when the piston is at the end of its travel or approaching it. Whereas with the link motion, either by derangement or excessive wear, the valve laps the ports at the end of the stroke, thus causing excessive compression and many other troubles.

Another feature of the motion which appeals to the engineer is the ease of handling the reverse lever when the locomotive is running at a high rate of speed.

Table 46.—General Dimensions.

Freight Locomotive—Mallet Compound, with Walschaert Valve Gear Built for Baltimore & Ohio Railroad by the American Locomotive Company.

Gauge of track, 4 feet 8½ inches.

Loaded Weights.

On driving wheels	334,500 pounds
Total engine	334,500 pounds
Tender	143,000 pounds

Wheel Base.

Driving	10 feet and 10 feet
Total of engine	30 feet 8 inches
Total of engine and tender	64 feet 7 inches

Cylinders.

Diameter.....20 inches and 32 inches
Stroke of piston.....32 inches
Valves.....H. P. piston, L. P. slide

Wheels.

Diameter of driving wheels, outside.....56 inches
Diameter of tender wheels.....33 inches

Journals—Diameter and Length.

Driving .....9x13 inches
Tender .....5½x10 inches

Type, 0660 C 334, Mallet Articulated

Boiler.

Type.....Straight Top, Radial Stay
Outside diameter at front end..... 84 inches
Length of fire-box, inside.....108 inches
Width of fire-box, inside..... 96 inches
Number of tubes.....436
Diameter of tubes.....2¼ inches
Length of tubes.....21 feet
Working pressure per square inch.....235 pounds
Heating surface in tubes.....5,366.3 sq. feet
Heating surface in fire-box..... 219.4 sq. feet
Total heating surface.....5,585.7 sq. feet
Grate area ..... 72.2 sq. feet

Tender Capacity.

Water .....7,000 gallons
Fuel .....15 tons

Clearance Limitations.

Height of stack above rail.....15 feet
Width.....10 feet 5 inches
Length over all.....79 feet 6½ inches

Maximum tractive power, 71,500 pounds working compound.

DETAILS OF CONSTRUCTION.

Having presented a general description of the Walschaert gear, together with a few of the advantages to be derived from its use, it is now in order to consider it more in detail. Valve gear has not improved proportionately to the improvement of other factors in connection with locomotive development during the past twenty-five years, and it is certainly to the interest of engine men, machinists, engineers, and all others connected with the construction, maintenance, and operation of locomotives, to familiarize themselves with a valve gear that offers so many mechanical and structural advantages over the Stephenson motion as does the Walschaert gear. The facts are that this valve gear has become almost a necessity in modern locomotive practice, owing to conditions created by the demands of transportation.

The Walschaert valve gear has long been in use on the main line locomotives in Europe, about 90 per cent of the engines there being equipped with it, and American designers now turn to this gear to enable them to meet present and future conditions. The principal reasons given by the builders of American locomotives for applying the Walschaert gear to their engines, especially those of the larger types, may be summed up as follows:

(1) *Accessibility.* There is not room enough for the Stephenson gear under a very large passenger or freight locomotive. The eccentrics are crowded, and proper inspection, not to speak of proper care, is difficult, except over a pit. Valve gear to be properly maintained must be accessible for inspection and lubrication. The accessibility of Walschaert gear should reduce engine failures.

(2) *Weight.* A saving of 1,745 pounds is possible by using the Walschaert gear, in the case of a very heavy passenger locomotive (see table 47). Stephenson gear, weighing as much as two tons, is far too heavy to be satisfactorily reversed twice in every revolution on fast running locomotives.

(3) *Directness.* Walschaert gear transmits the moving force to the valve in very nearly straight lines, avoiding the springing and yielding of the rocker arms, rocker shafts and transmission bars, which cannot be avoided in these parts of the Stephenson motion, even if they are made very heavy.

(4) *Permanency of Adjustment.* The advantage of permanence of adjustment lies with the valve gear which has no large eccentrics. All connections in the Walschaert gears are made with pins and bushings, which are designed specially to resist wear.

(5) *Wear.* Large eccentrics, besides occupying too large space, wear unevenly, and lubrication is difficult with the high surface velocities of the largest sizes. With hardened pins and hardened bushings the Walschaert gear has an important advantage in maintenance.

(6) *Smooth Operation.* Stephenson links, under the influence of two eccentrics, move through wide angles, resulting in a wedging action of the link block, which strains the gear when working hard, and produces lost

motion. Walschaert links oscillate through smaller angles, producing less lost motion. The effect of this angularity of the links is plainly discernible on the testing plant.

(7) *Frame Bracing.* The removal of the valve gear from between the driving wheels facilitates bracing the frames of the locomotive laterally.

The great saving in weight of the various parts of the Walschaert valve gear, as compared with the Stephenson link motion, is shown in table 47.

It may be assumed that the student is familiar with the meaning of the terms lap, lead, and cut-off, when used in connection with the operation of the valve, also that the principles governing the action of outside admission valves, as compared with those of inside admission, are thoroughly well understood; therefore it will not be necessary to occupy space by an explanation of these matters.

In the construction of the Walschaert gear the desired travel of the valve, the lead and the maximum cut-off which determines the lap of the valve, are selected. The stroke of the piston being given, the combination lever is proportioned so that a motion equal to the lap and lead is given to the valve when the crosshead is moved from one end of the stroke to the other. The link may be made of any approved design, and is so located that the radius bar will have a length of at least eight times (ten or twelve times is better) the travel of the link block, and the radius of the link should be equal to the length of the radius bar.

For outside admission valves the radius bar is attached to the combination lever between the valve stem and the crosshead connections, and for inside admission (piston

TABLE 47.—(AMERICAN LOCOMOTIVE CO.).

Saving in Weight with Walschaert Valve Gear—Comparative Weights, Stephenson and Walschaert Valve Gear—Lake Shore & Michigan Southern Locomotive.

Crank pins, main	520	490	390	365	440	415
Crank pin arms	...	100	...	80	...	90
Crosshead arms	...	60	...	50	...	50
Eccentric	600	...	740	...	740	...
Eccentric strap	800	...	880	...	1,120	...
Eccentric rods	200	220	200	175	280	200
Link	280	260	260	240	300	275
Link support	...	280	...	250	...	260
Link lifter	45	...	120	...	120	...
Reverse shaft and arms	260	400	350	375	385	390
Rockers	260	...	280	325	280	350
Rocker boxes	240	...	300	300	300	325
Transmission bar	300	140	270	160	300	170
Transmission bar hanger	80	72	120	65	120	75
Transmission bar bracket	...	...	...	...	200	...
Valve rod	80	70	130	100	100	100
Vibrating rod	...	220	...	180	...	180
Vibrating link	...	70	...	60	...	60
Total, lbs.	3,666	2,382	4,040	2,725	4,685	2,940
Saving in weight, lbs.		1,283		1,315		1,745

valves) it is attached above the valve stem. The fulcrum of the link should lie as nearly as practicable upon a line drawn through the union of the radius bar and combination lever, parallel with the center line of the valve stem. The suspension point of the lifter should have a locus which causes the link block to travel as nearly as practicable on a chord of the arc described by any point of the link wherever the block happens to be when the link is swung into one of its extreme positions. This is most closely approached by a lifter through which the radius bar slides, and does not swing with the link. A properly suspended hanger will accomplish practically the same result, though the slip of the link bar will be somewhat more in the back than in the forward motion, but as the suspension point cannot be made to follow the theoretical locus, it should be made to do so as nearly as possible by favoring the position of the most commonly used cut-offs. In locating the longitudinal position of the link fulcrum, consideration should also be given to the length of the eccentric rod, which should have a minimum length of three and one-half times the eccentric throw, and should be made as long as circumstances will permit, with an approximately equal length of the radius and eccentric rods. The point of connection between the eccentric rod and the link should be as near the center line of motion of the main rod as this correction for rod angularities will permit, but this is often accompanied with the requirement of excessive eccentric throw. In such cases a compromise must be made to raise this point. The fore and aft position of this point relative to the tangent of the link arc must also be determined with reference to the angularity of the eccentric and main rods, so that the link is exactly in its central position when the piston is at either

end of the stroke. The angles through which the link swings on both sides of its central position should be as nearly as practicable equal, but this is subordinate to other conditions. Attention should be paid to the effect of the angularity of the main connecting rod upon the cut-off, to reduce this to a minimum, this having an effect upon determining the locus of the suspension point of the lifting link as well as that of the eccentric rod connection to the link.

Walschaert gears should be correctly laid out and constructed from a diagram, as the proportions cannot be tampered with by experimental changes without seriously affecting the correct working of the device.

The only part capable of variation in length is the eccentric rod, which connects the eccentric with the link. This rod may be slightly lengthened or shortened, to correct errors in location of the link center, from center of driving axle which carries the eccentric.

The eccentric usually assumes the form of a return crank on one of the crank pins, and its center is at right angles to the plane of motion, viz. :—at ninety degrees to a line drawn from the point on the link at which the eccentric rod is attached, through the center of the driving axle. This eliminates the angular advance of the eccentric, and allows the use of a single eccentric for both forward and backward motion. The throw as specified must be correctly obtained, and great care taken that the position shown in the design be adhered to. The crank representing the eccentric is permanently fixed to the pin, and the slightest variation will be detrimental.

METHOD OF LAYING OUT WALSCHAERT GEAR.

Having presented a general outline of the gear, we may proceed in determining the more important points necessary to obtain a successful motion of the valve, and, as previously stated, the stroke of the engine is given, and the travel and lap and lead of the valve are selected to suit a desired cut-off. We have first to find the proportions of the combination lever. By designating the lap and lead with a letter C, the crank Radius with R, the crosshead end of the combination lever with *L*, and the valve end of the same with V, we have R:C = L: V, or $V = \frac{CL}{R}$, with the connection F of the radius bar as a fulcrum. The length of the combination lever must be determined from a height of the valve stem over the piston rod and a convenient angle of oscillation of 45° to 50°, which should not exceed 60°.

We have next to find the required travel of point F, Fig. 16, to obtain the desired valve travel, which for convenience sake is taken on one side of the center position, or half its total travel in full gear, and which we will designate b, when we have:

$$b = \frac{R\sqrt{a^2 - c^2}}{R+c} \quad \text{for outside admission, and}$$

$$b = \frac{R\sqrt{a^2 - c^2}}{R-c} \quad \text{for inside admission valves,}$$

This may be laid out graphically as in Figs. 17 and 18, when *a* is equal to one-half the travel of the valve and R and *c* the same as in the above formulæ.

With the limited amount it is advisable to allow in

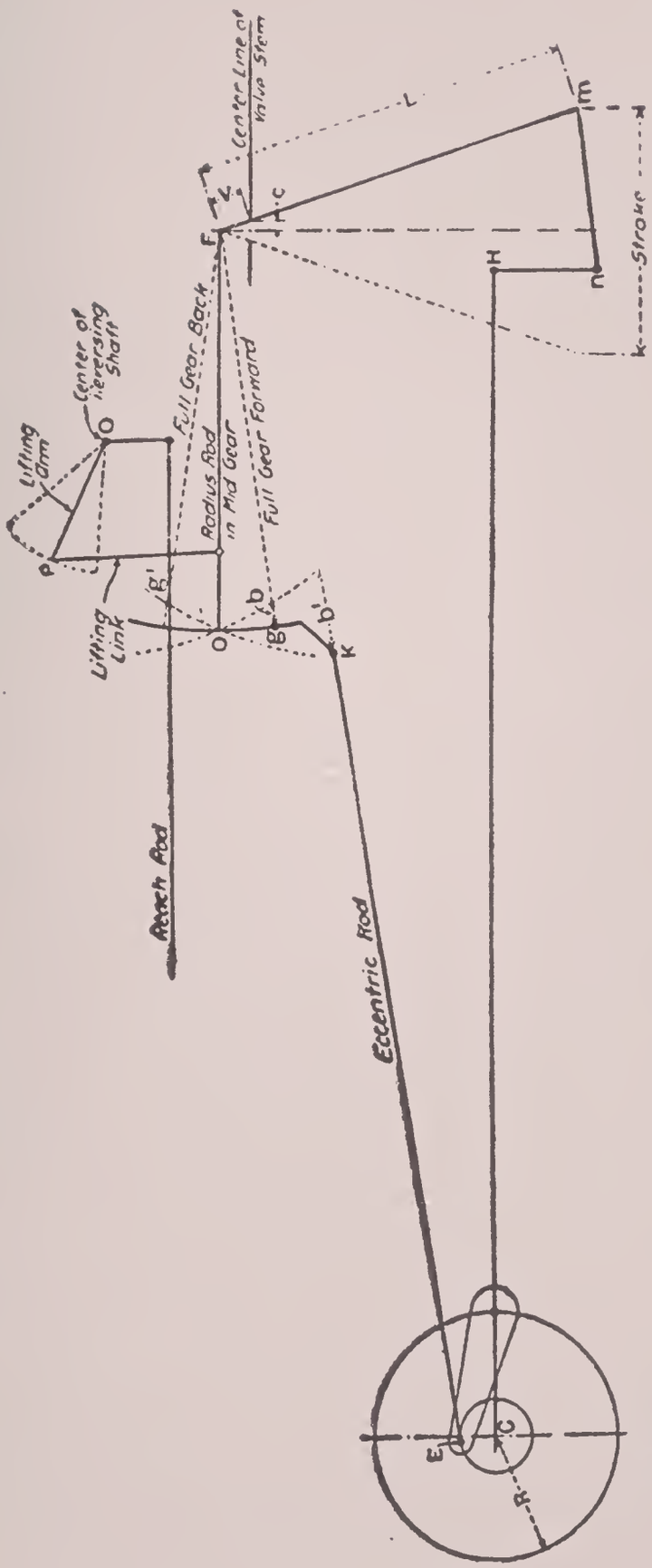


FIGURE 16. DIAGRAM ILLUSTRATING METHOD OF LAYING OUT THE WALSCHAERT VALVE GEAR.

raising or lowering the link block in reversing the motion, we can without practical error consider the half movement of the link block g to be the same as that of point F, and by limiting the angle of the swing of the link to a maximum of 45°, we get the raise or depression of the radius bar and link block $Og = \frac{b}{\text{tan. } d}$, where O is the link fulcrum and d = half the angle of the swing of the link.

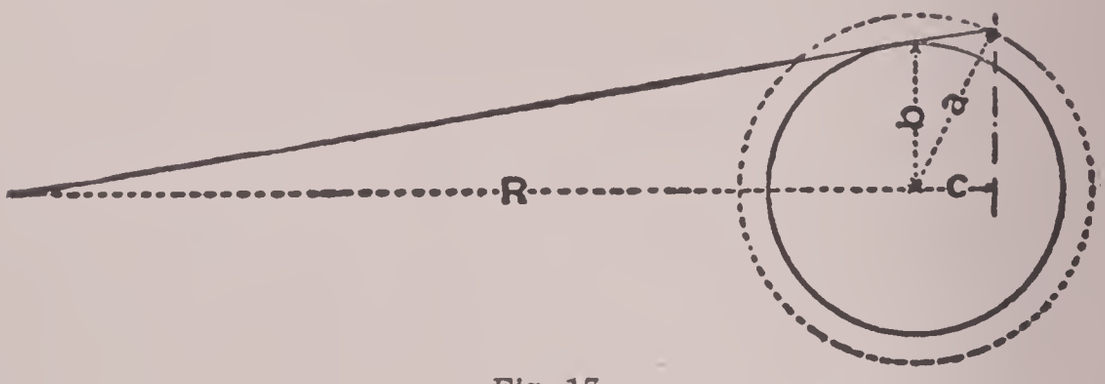


Fig. 17.

The location of the link and eccentric rod connecting point K, Fig. 16, cannot be determined with any practicable formula, but must, as already stated, be found by plotting to meet the requirements of the different cut-offs and corresponding crank positions. The same is also the case with determining the locus for the suspension point P of the lifting link, and in these two locations lies the principal success of the gear.

NOTE.—The cuts, Figures 16, 17 and 18, are reproduced from an article by Messrs. C. J. Mellin and G. L. Fowler in *Railway Machinery,* September, 1905.

Figure 19 illustrates the Walschaert valve gear as applied to a heavy freight locomotive, in fact one of the largest type belonging to the Lake Shore and Michigan Southern Railway.

This cut will serve to show in a graphic manner the location of the various connections of the valve gear, and piston valve. The combination lever is shown in its central position and it will be noticed that in this position the lower connection, m (see Fig. 16), is in line horizontally with point n, the crosshead connection. The vertical height of these two points relative to each other slightly influences the port opening. The cross section through cylinder and valve shown in Fig. 19 will give a clear idea of the action of the valve and gear.

Figure 20 is an outline diagram of the Walschaert valve gear as applied to a compound locomotive, and serves to show the adaptability of the gear to any type of locomotive and all conditions of service.

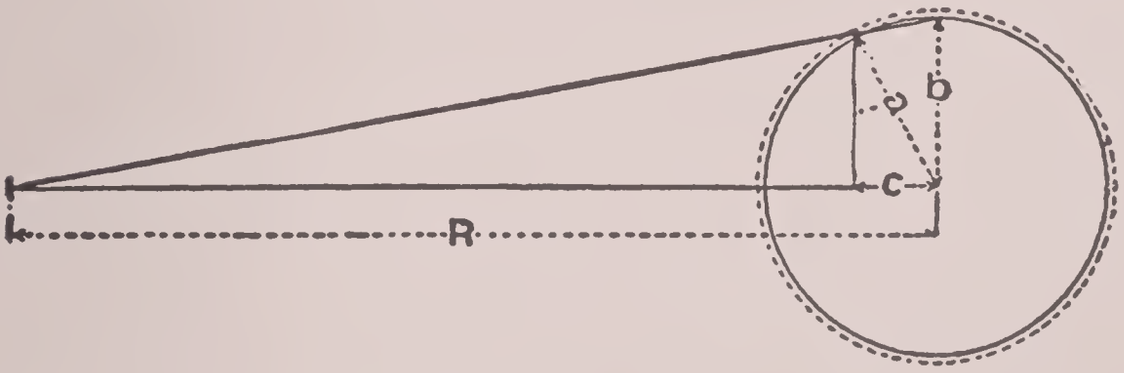


Fig. 18.

But, it is evident that a correct design of the Walschaert valve gear can only be laid out by a skilled draughtsman, and in the adjustment and maintenance of the gear the greatest care is required, that all parts should preserve their original forms and locations, and this should be checked by verifying the valve events, through turning the main driving wheels before the locomotive goes into active service.

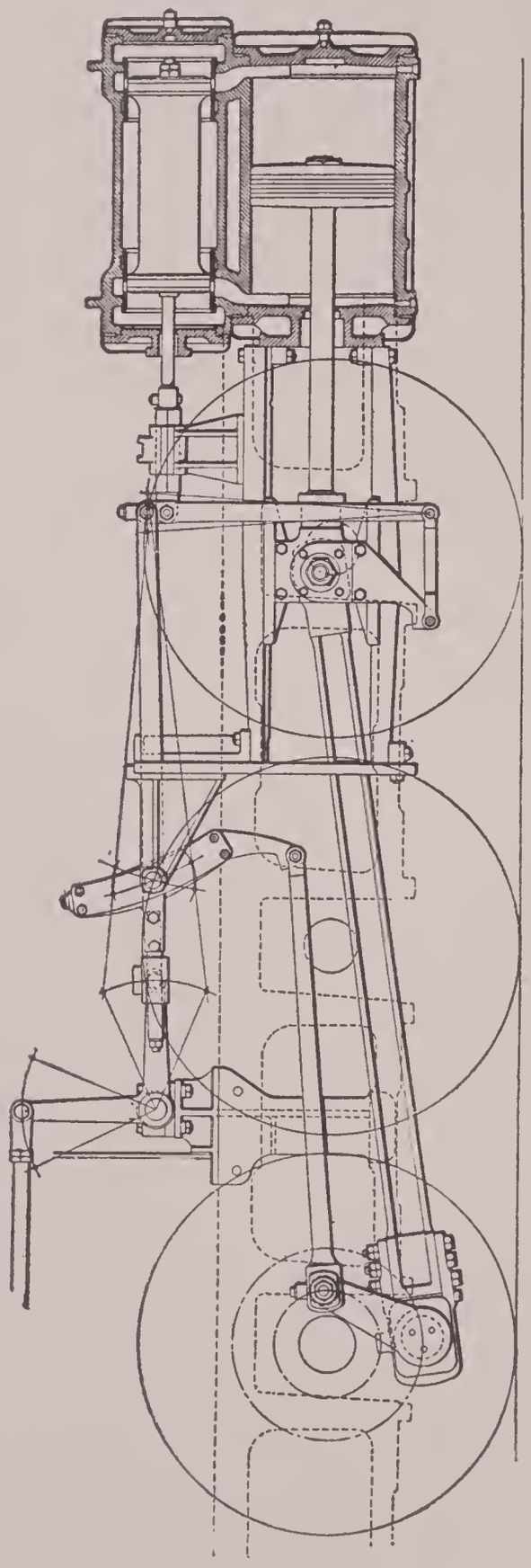


FIGURE 19. WALSCHAERT VALVE GEAR.

As applied to Consolidation Freight Locomotive, Lake Shore & Michigan Southern Railway, illustrating a cross section through cylinder and valve—American Locomotive Company.

The Walschaert motion, as usually constructed, does not lend itself as freely to adjustment as does the Stephenson motion with independent eccentrics, and for this reason it is not as liable to get out of adjustment. It must be correctly laid out in the design and correctly fitted up. The importance of this cannot be overestimated. The various points must be carefully plotted in order to give the best results in the combination movement of the parts of the motion. The movements of the motion involve such complications in plotting as to render the complete plotting of all, too laborious for every new design, and for this reason the use of an adjustable model is very desirable in designing this gear. However, with complete knowledge of the nature of the gear, simple methods and formulæ may be used to determine the locations of the various points covering the motion. One object of this description is to avoid the necessity of a model except to verify the results.

To entirely overcome the irregularities inherent in all motions transformed from circular into lineal, cannot for practical reasons be expected, but the errors may be reduced to such an extent that they do not affect either the power or economy of the locomotive. This remark is made to forestall the inference that the accuracy of the Walschaert motion as to the cut-off points is not superior to the Stephenson motion when the latter is turned out of the shop.

The chief point of difference between the Walschaert and Stephenson motions is that the former gives to the valve a constant lead at all cut-offs, whereas the latter produces an increase of lead which becomes excessive at short cut-offs.

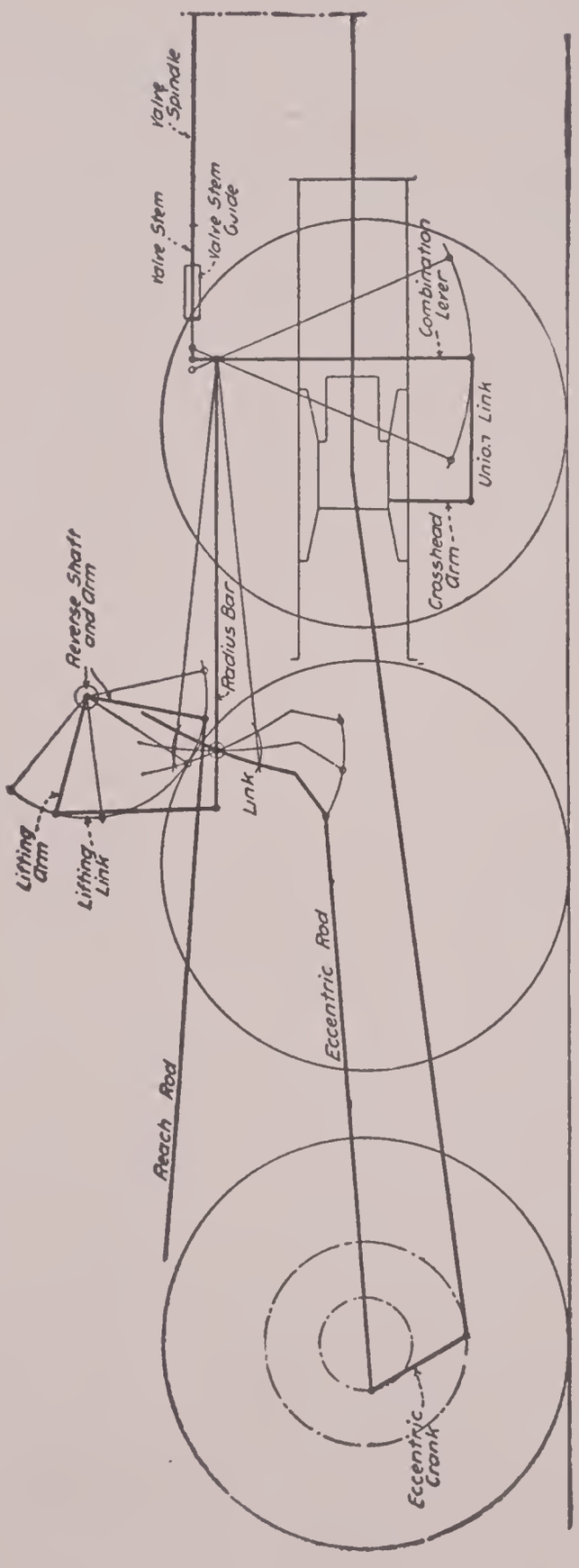


FIGURE 20. OUTLINE DIAGRAM OF WALSCHAERT VALVE GEAR.
As applied to Mallet Articulated Compound. Baltimore & Ohio Railroad.

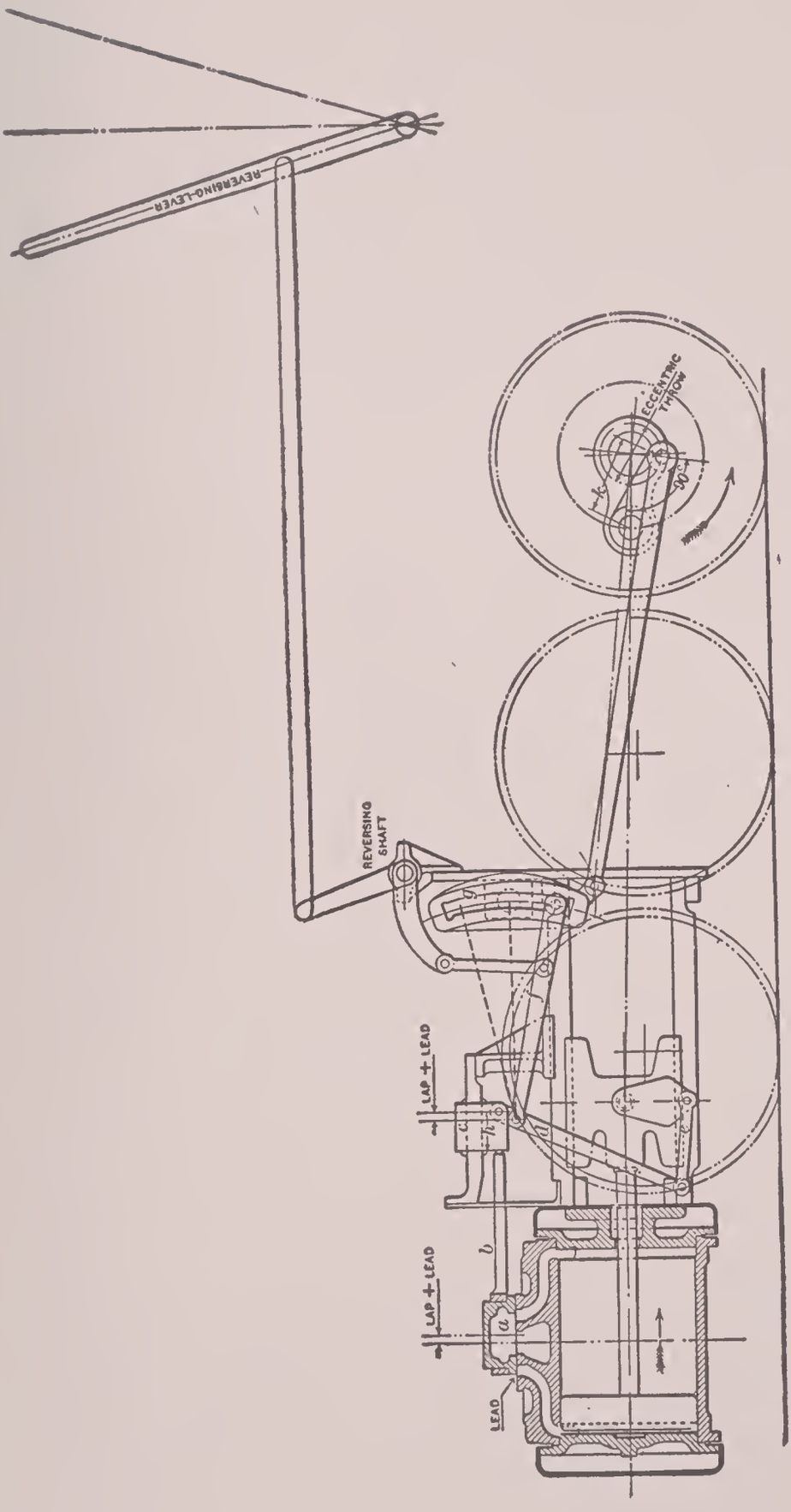


FIGURE 21. WALSCHAERT VALVE GEAR DIAGRAM.

DETAILS OF ADJUSTMENT.

In setting the Walschaert valve gear it must be borne in mind that two distinct motions are in combination, viz.: the motion due to the crosshead travel, and the motion due to the eccentric throw.

The crosshead motion controls the lead, by moving the valve sufficiently to overcome its lap, by the amount of lead in both front and back positions. The eccentric throw controls the travel and reversing operations. It will be seen that the movement due to the eccentric, without the crosshead motion, would place the valve centrally over the ports when the piston is at the extreme end of the stroke. The combined effect of these two motions, when the parts are properly designed, gives the required movement of the valve, similar to that obtained by the use of a stationary link. To reverse the engine, the link block is moved from end to end of the link, instead of moving the link on the block. This operation is accomplished by means of a reversing shaft connected with a reversing lever in the cab.

When the engine is assembled, the throw of the eccentric should be checked up by the specifications, and any error should be reported at once, in order that the mistake may be rectified by either correcting the position of the eccentric, or by a change in the design of the other parts to compensate for the error.

The lap and lead being determined by the proportion of the arms of combination lever, and the stroke of the piston, the amount is found by turning the engine from one dead center to the other in any cut-off position.

Mr. Carl J. Mellin of the American Locomotive Company furnishes six very useful hints relating to the adjustment of the Walschaert valve gear as follows:

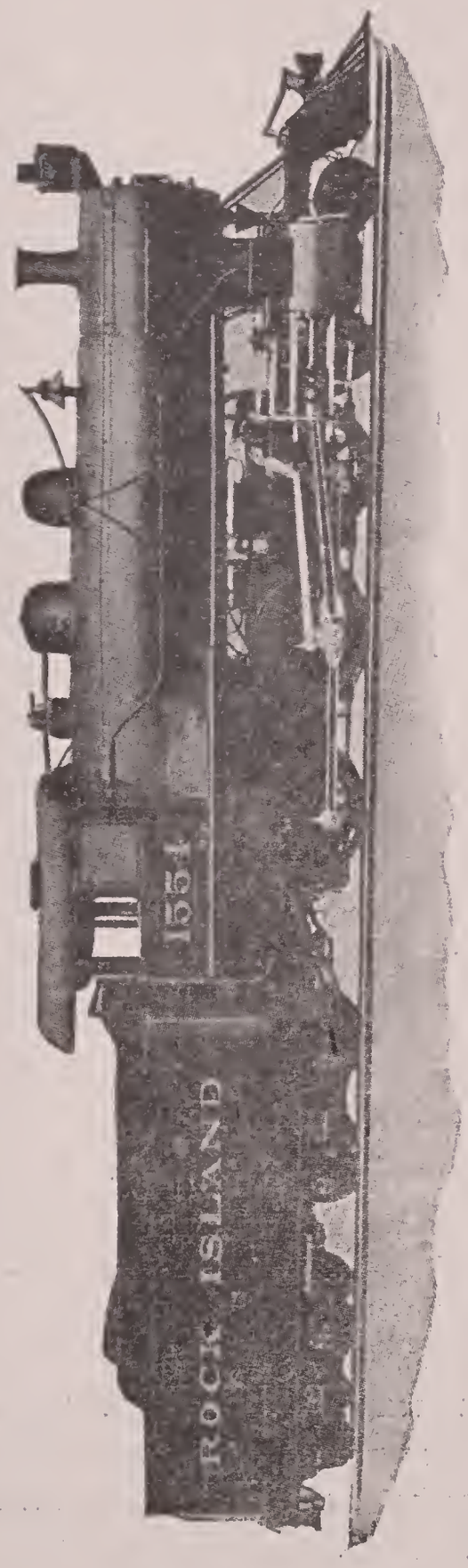


FIGURE 22 TEN WHEEL LOCOMOTIVE FOR THE CHICAGO, ROCK ISLAND & PACIFIC RAILWAY, 1905.
Equipped with Walschaert Valve Gear. (Baldwin Locomotive Works.)

1. The motion must be adjusted with the cranks on the dead centers by lengthening or shortening the eccentric rods until the link takes such a position as to impart no motion to the valve when the link block is moved from its extreme forward to its extreme backward position. Before this change in the eccentric rod is resorted to, the length of the valve stem should be examined, as it may be of advantage to plane off, or line under, the foot of the link support which might correct the lengths of both rods or at least only one of these should need to be changed.

2. The difference between the two positions of the valve on the forward and back centers is the lap and lead doubled, and cannot be changed except by changing the leverage relations of the combination lever.

3. A given lead determines the lap, or a given lap determines the lead, and it must be divided for both ends as desired by lengthening or shortening the valve spindle.

4. Within certain limits this adjustment may be made by shortening or lengthening the radius bar, but it is desirable to keep the length of this bar equal to the radius of the links, in order to meet the requirements of the first condition.

5. The lead may be increased by reducing the lap, and the cut-off point will then be slightly advanced. Increasing the lap introduces the opposite effect on the cut-off. With good judgment these quantities may be varied to offset their irregularities inherent in transforming rotary into lineal motions.

6. Slight variations may be made in the cut-off points as covered by the previous paragraph, but an independent adjustment cannot be made except by shifting the location of the suspension point, which is preferably determined by a model.

The Baldwin Locomotive Company supplies the following "Special Instructions" for guidance in the erection of the Walschaert valve gear and setting the valves:

1. Check carefully the dimensions of the following parts, rejecting any that are not exactly to drawing:

a—Valve
b—Valve stem.
c—Valve crosshead or slide.
d—Combining lever.
e—Crosshead link.
f—Link radius rod.
g—Reverse link
h—Location of combining lever on crosshead.
k—Length of eccentric crank.

2. Check eccentric throw to see that it is exactly as specified.

3. Be sure that guide bearer is correctly located from center of cylinder, as the reverse link is usually attached to it, and variations in the location of the link cannot be allowed. If the link is attached to separate crosstie, similar precautions must be taken to insure its correct location.

4. Exercise great care in the location of the link, so that the trunnion center is exactly to dimensions from the center of cylinder.

5. See that the reverse shaft center is correctly located to dimensions given, and that the lifting arm and link are of the exact lengths as specified.

6. Connect crosshead gear to valve, and radius rod to link, without connecting eccentric rod to link.

7. Hook up radius rod to exact center of link, and then revolve driving wheels, seeing that crosshead gear

gives correct lead as specified for both front and back admission ports.

8. Connect link to return crank by eccentric rod, and obtain full travel front and back, and in both forward and backward motions, correcting any errors by lengthening or shortening eccentric rod, as previously noted.

The valves may now be considered as definitely set, and may be tested to any cut-off points in the usual manner. A simple additional check should be made as follows: set one side of the engine so that the piston is at its extreme forward position in the cylinder, and check the lead on the admission port.

In this position it should be possible to move the link block through its entire travel in the link, without in any way disturbing the movement of the valve.

This operation should then be reversed, and the other side of the engine similarly tried with the piston located at its extreme backward position in the cylinder.

QUESTIONS.

58. What is the chief difference between the Walschaert, and the Stephenson valve gears?

59. How is the Walschaert gear operated?

60. How is the link secured?

61. Describe in general terms the construction, and operation of the radius rod and link.

62. Is a rock-shaft always necessary with the Walschaert valve gear?

63. What is the function of the combination lever?

64. Describe the connection, and operation of this lever.

65. Where is the fulcrum of the combination lever with outside admission valves?

66. With inside admission valves where is the fulcrum located?

67. What should the radius of the link be equal to?

68. How can this be ascertained?

69. What are the two essential points to be kept in view in the design and construction of the Walschaert valve gear?

70. If, after connecting the gear there should be a slight variation between the forward and backward position of the valve, how may it be corrected?

71. What disadvantage attends the investigation of the Stephenson valve gear on a modern locomotive when out on the road?

72. How many eccentrics, or eccentric cranks are required to drive the Walschaert valve gear?

73. How does this gear compare with the Stephenson gear, in the items of size of bearings, and friction?

74. Does the lead of the valve vary at different points of cut-off with the Walschaert gear?

75. Mention some of the advantages to be gained by a constant lead.

76. Is the Walschaert valve gear as easy to handle from the cab as is the Stephenson?

77. Has the valve gear of locomotives kept pace with other parts in general development and improvement?

78. Owing to the demands of modern transportation, what can be said of the Walschaert valve gear?

79. What are its advantages over the Stephenson gear as regards accessibility?

80. What is the saving in weight effected by the use of the Walschaert gear?

81. What can be said of this gear as regards directness of transmission?

82. As to permanence of adjustment?

83. How does it compare with the Stephenson gear in the matter of wear?

84. In regard to smooth operation what can be said of the Walschaert gear?

85. How is the frame bracing affected by the use of this valve gear?

86. In the construction of the Walschaert valve gear how are the proportions of the combination lever determined?

87. What factors govern the location of the link?

88. What rules are to be observed in the location of the suspension point of the lifter?

89. In the location of the longitudinal position of the link fulcrum, what points are to be considered?

90. Where should the point of connection between the eccentric rod and the link be located?

91. If this location should cause too much eccentric throw, how may a compromise be affected?

92. What important rule should govern the laying out, and construction of the Walschaert valve gear?

93. Why should this rule be strictly observed?

94. Is there any part of this gear in which changes may be made, once it is erected?

95. What form does the eccentric usually assume in the Walschaert valve gear?

96. What is the position of the center of the eccentric crank relative to the plane of motion?

97. Has the eccentric an angular advance?

98. Does the Walschaert valve gear lend itself as freely to adjustment as does the Stephenson link?

99. What great advantage has the Walschaert gear over the Stephenson gear as regards the lead given the valve?

100. What motion controls the lead?

101. What motion controls the valve travel and reversing operations?

102. Where would the motion of the eccentric alone place the valve when the piston is at the end of the stroke?

103. Wherein does the reversing operation of the Walschaert valve gear differ from that of the Stephenson link?

104. What precaution should be observed regarding the position and throw of the eccentric when the engine is assembled?

105. How is the amount of lap and lead ascertained?

106. In the adjustment of this valve gear, how is the length of the eccentric rod determined?

107. What should be done with the valve stem before changing the eccentric rod?

108. What does the difference between the two positions of the valve on the forward and back centers represent?

109. Can this be changed?

110. How is the lead equalized for both ends?

111. How may the lead be increased?

112. Referring to Special Instructions by the Baldwin Locomotive Co. regarding the erection of the Walschaert valve gear, what precautions are to be taken as to the dimensions of the various parts?

113. What should be done with the eccentric?

114. What suggestion is made concerning the guide bearer?

115. What should be done regarding the location of the link?

116. What is said concerning the reverse shaft, lifting arm and link?

117. Concerning connections of crosshead gear, or combination lever, and the radius rod to the valve, what method should be pursued?

118. If any errors exist after the eccentric has been connected, how may they be corrected?

119. After the valves have been adjusted, in common parlance, "set," how may they be tested?

120. Referring to types of valves, what is the principal objection to the use of the D slide valve?

121. Is the piston valve a perfectly balanced valve?

122. What is a necessary factor in order that a valve may be perfectly balanced?

123. Can an inside admission valve be balanced similar to an outside admission?

124. Why should a piston valve be made as long as possible?

125. Is there any particular advantage in the use of inside admission valves over those of outside admission?

126. Name four simple rules by which an engineer may ascertain whether the valves of his engine are inside or outside admission.

127. Mention a very common defect of snap-ring piston valves.

128. Describe in general terms the Wilson high pressure valve,

129. Is the balanced area of this valve the same at all points of its travel?

130. Does the port pressure influence the balancing of the Wilson valve?

131. Mention the names of five types of balanced valves for locomotives that are being used extensively.

132. How may the actions of the valve and the relative positions of the crank pin and eccentric be illustrated?

133. What diagrams are generally used for this purpose?

134. Explain the reason for the increase of lead as the reverse lever is hooked back with the Stephenson valve gear.

135. Upon what does the amount of this increase of lead depend?

136. Why is there no change in the lead with the Walschaert valve gear?

137. What part of this gear is it which forms the fulcrum for the combination lever?

138. What is the first requisite to be considered in connection with this gear?

139. What is the second requisite in connection with the operation of the Walschaert valve gear?

140. What is implied in the third requisite?

141. Does the vibration of the engine upon its springs disturb the Walschaert valve gear in its operation?

142. What precautions are to be observed in order to attain this object?

143. When the piston is at mid stroke, is the crank pin center exactly perpendicular above, or below the center of the driving axle?

144. What then is the position of the crank with the piston at mid stroke, and what causes it to assume that position?

145. Where will the center of the crosshead be with the piston at midstroke?

146. What should be the position of the combination lever of the Walschaert gear, with the piston at mid stroke?

147. How can this be ascertained?

148. What should be the position of the link fulcrum pin, and the link block pin when the reverse lever is in the central notch of the quadrant?

149. If these pins are not in their correct position, what is the probable cause of the error?

150. How may an off hand test of the valve adjustment be made? Describe the process.

151. If by this test the position of the valve is found to be wrong, how may it be righted?

152. What precaution should be observed in regard to making changes in this gear after it has been turned out of the back shop?

153. Describe the method of correcting the length of the eccentric rods.

154. Mention a very important feature in connection with the link support.

155. What other portion of this gear permits of no variation of distances?

156. What can be said of the Walschaert valve gear regarding its source of motion, and the location of its eccentric crank?

157. Does this apply to the Stephenson gear in all cases?

158. What especial type of engine would be totally disabled by the breaking of a side rod if equipped with the Stephenson link?

159. Why should the engine be disabled?

160. If such an engine was equipped with the Wal-

schaert gear would such an accident disable it from proceeding further?

161. Give the reason for this.

162. In case of an accident to the eccentric crank, or eccentric rod of a Walschaert gear, what is the proper method to pursue?

163. If a breakdown of this kind should occur with the engine having but a short distance to cover to reach the terminal station, what would be a quick method of solving the problem?

164. Is this method to be recommended at all times?

165. Describe a much safer method which will permit the engine to be speeded up with one side after the job is completed.

166. Is it necessary in all cases to disconnect the combination lever?

167. What precaution must be observed in this particular?

168. What is a good plan to pursue in the way of preparation for breakdowns?

169. What particular feature must not be neglected with any kind of a breakdown when it is desired to proceed with one side?

170. What should be done in case of a broken radius bar?

171. Is the lifting arm, or suspension bar of the Walschaert valve gear always located in the same position relative to the link?

172. In case the lifting arm should break, or the extension of the radius bar back of the link should become bent or broken while the connection of the radius bar to the link remained intact, what method should be pursued?

173. When temporary repairs of this kind are made what precaution regarding the reverse lever must be observed?

174. When a valve stem breaks what may be done to fit the engine for running with one side?

175. If the combination lever or its connection to the crosshead gets bent or broken what should be done?

176. What must always be done with the valve, on the disabled side?

177. In case of a broken main rod what is the proper course of procedure?

178. If the valve is an inside admission valve, what must be done with it?

179. If the engine has outside admission valves what must be done with the valve?

180. What is the object aimed at in thus securing the valve?

181. What must be done with the crosshead in case of a broken main rod?

182. Why is this precaution necessary with the crosshead?

183. What must be done with a broken or disconnected piston?

184. Is it necessary to take down the main rod in all cases of a broken piston?

185. When is it absolutely necessary to take down the main rod for a broken piston?

186. If a front cylinder head should get broken or blown out what should be done?

187. Why should the back cylinder cock be removed in this case?

188. In case a back cylinder head should get broken or blown out what should be done?

189. If the break in the back cylinder head is not serious what may be done?

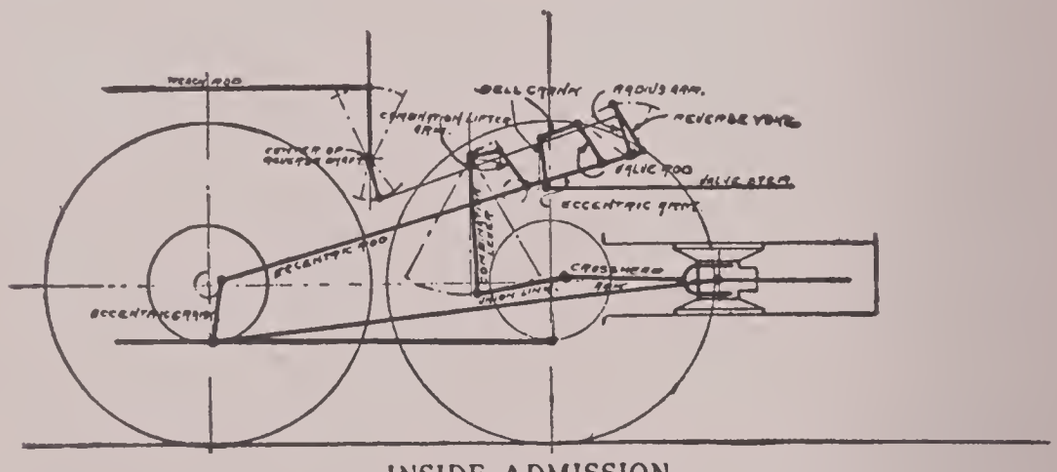
190. What is the proper method of procedure when the back cylinder head is badly broken?

191. If the connecting bar or link between the cross-head and the lower end of the combination lever should happen to get broken and lost from both sides of the engine, how may the lengths of the lost bars be obtained?

THE BAKER-PILLIOD VALVE GEAR.

Figures 23 and 24 illustrate the Baker-Pilliod valve gear for inside, and outside admission, and Figs. 25 and 26 show the general arrangement.

With this valve gear the motion of the valve is also derived from two independent sources, viz.: the crank pin, through an eccentric crank set at right angles to the main crank, and connection to the crosshead by means



INSIDE ADMISSION

Fig. 23. Baker-Pilliod Valve Gear.

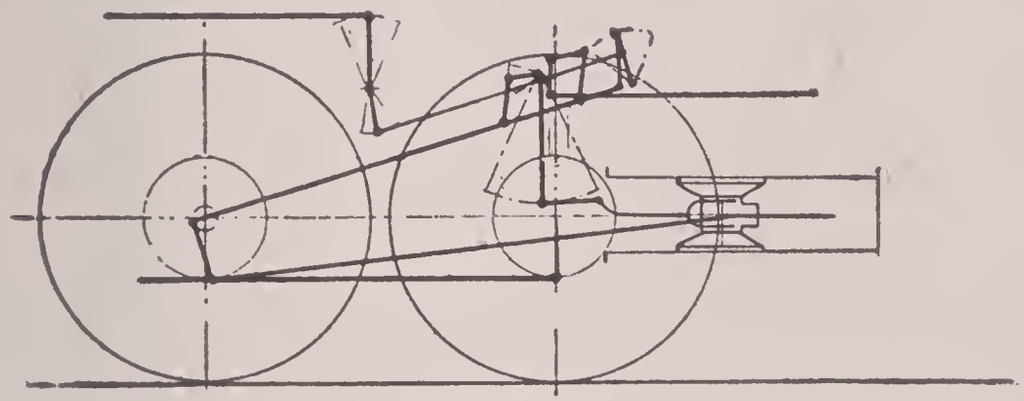
of a swinging lever, similar to the Walschaert valve gear.

The swinging lever, or radius bar is suspended from a reversing yoke, that may be moved to any desired angle to impart the required throw and cut-off.

The lever action of the eccentric arm actuates the lap and lead and maintains a constant lead.

The cross-head connection imparts a motion of lap and lead when the swinging lever, or radius arm, and the reversing yoke are in its central position.

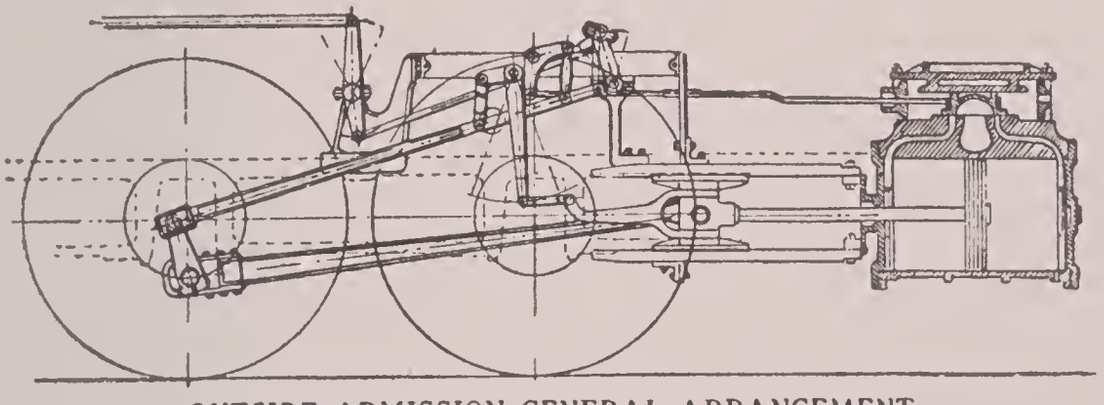
Therefore in mid-gear, with the reverse lever in the center notch, this will be practically all the motion imparted to the valve.



OUTSIDE ADMISSION

Fig. 24. Baker-Pilliod Valve Gear.

By moving the reverse lever forward the angle of the reverse yoke is changed and brought into combination with the main imparting motion towards the eccentric arm, while the opening motion of the valve is accelerated for the forward motion of the engine.



OUTSIDE ADMISSION GENERAL ARRANGEMENT

Fig. 25. Baker-Pilliod Valve Gear

For the backward movement, the reverse yoke is changed to an opposite position, while the path of the valve rod remains the same as for the go-ahead movement.

Fig. 26. Baker-Pilliod Valve Gear.

Table 48.—Valve Motion Report. T. St. L. & W. R.R. Eng. 42, Taken May 20, '06. 10-Wheel, Baldwin Passenger Eng. Baker-Pilliod Valve Gear. 19"x24" Cylinders, 69" Drivers, Valve Travel 4¾", Steam Lap 1 1/16", Lead Constant 3/16".

CUT OFF POSITION.			PRE-ADMISSION.		PORT OPENING.		CUT OF.		RELEASE.		COMPRESSION.		LEAD.	
			Frt. In.	Bck. In.	Frt. Inches.	Back. Inches.	Frt. Inches.	Back. Inches.	Frt. Inches.	Back. Inches.	Frt Inches.	Back. Inches.	Frt. In.	Bck. In.
Forward Motion	Per cent.	Full Gear	0	0	1¼	1¼	18¼	18¼	23	23	1	1	3/16 Constant	3/16 Constant
	66		0	0	1 3/16	1 3/16	16	16	22¾	22¾	1¼	1¼		
	50	Half Stroke	0	0	⅝	⅝	12	12	22½	22½	1½	1½		
	33		0	0	7/16	7/16	8	8	21½	21½	2½	2½		
	25	Quarter Stroke	0	0	5/16	5/16	6	6	20¾	20¾	3¼	3¼		
	20		¼	¼	5/16	5/16	4¾	4¾	20¼	20¼	3¾	3¾		
Backward Motion		Full Gear	0	0	1⅛	1¼	17¾	18½	23	23	1	1	3/16 Constant	3/16 Constant
	66		0	0	1 3/16	1	16	16¾	22¾	22½	1¼	1½		
	50	Half Stroke	0	0	⅝	¾	12	12	22½	22½	1½	1½		
	33		0	0	7/16	½	8	8⅜	21	21½	3	2½		
	25	Quarter Stroke	0	0	5/16	5/16	6	6¼	20¾	21¼	3¼	2¾		
	20		¼	¼	5/16	5/16	4¾	5	20¼	20¼	3¾	3¾		

Table 49.—Valve Motion Report, A. L. Co., Brooks, Jan. 7, 1906. 10-Wheel Freight Eng. Stephenson Link. 19"x26" Cyl., 57" Drivers, Valve Travel 5", Steam Lap 7/8", Link Radius 38".

	CUT OFF POSITION.		PRE-ADMISSION.		PORT OPENING.		CUT OFF.		RELEASE.		COMPRESSION.		LEAD.	
			Frt. In.	Bck. In.	Frt. Inches.	Back. Inches.	Frt. Inches.	Back. Inches.	Frt. Inches.	Back. Inches.	Frt. Inches.	Back. Inches.	Frt. In.	Bck. In.
Forward Motion	Per cent.	Full Gear	0	0	$1\frac{11}{16}$	$1\frac{9}{16}$	23	23	$25\frac{3}{16}$	$25\frac{1}{4}$	$\frac{13}{16}$	$\frac{3}{4}$	0	0
	66		$\frac{1}{8}$	$\frac{1}{8}$	$\frac{3}{4}$	$2\frac{23}{32}$	$17\frac{3}{16}$	$17\frac{5}{16}$	$22\frac{7}{8}$	23	$3\frac{1}{8}$	3	$\frac{3}{16}$	$\frac{3}{16}$
	50	Half Stroke	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{15}{32}$	$12\frac{7}{16}$	$12\frac{3}{4}$	$20\frac{3}{4}$	$20\frac{7}{8}$	$5\frac{1}{4}$	$5\frac{1}{8}$	$\frac{1}{4}$	$\frac{7}{32}$
	33		$\frac{15}{16}$	$\frac{7}{8}$	$\frac{11}{32}$	$\frac{5}{16}$	$8\frac{3}{8}$	$8\frac{1}{2}$	$18\frac{9}{16}$	$18\frac{5}{8}$	$7\frac{7}{16}$	$7\frac{3}{8}$	$\frac{1}{4}$	$\frac{1}{4}$
	25	Quarter Stroke	$1\frac{3}{8}$	$1\frac{3}{8}$	$\frac{5}{16}$	$\frac{9}{32}$	$6\frac{3}{16}$	$6\frac{3}{16}$	$16\frac{11}{16}$	$16\frac{15}{16}$	$9\frac{5}{16}$	$9\frac{1}{16}$	$\frac{9}{32}$	$\frac{9}{32}$
	20		$1\frac{11}{16}$	$1\frac{13}{16}$	$\frac{19}{64}$	$\frac{17}{64}$	$5\frac{1}{8}$	$5\frac{1}{8}$	$15\frac{3}{8}$	$15\frac{3}{8}$	$10\frac{5}{8}$	$10\frac{5}{8}$	$\frac{9}{32}$	$\frac{9}{32}$
Backward Motion		Full Gear	0	0	$1\frac{5}{8}$	$1\frac{5}{8}$	$23\frac{1}{4}$	$23\frac{5}{16}$	$25\frac{1}{4}$	$25\frac{5}{16}$	$\frac{3}{4}$	$\frac{11}{16}$	0	0
	66													
	50	Half Stroke	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{1}{2}$	$12\frac{3}{8}$	$13\frac{3}{16}$	$20\frac{7}{8}$	21	$5\frac{1}{8}$	5	$\frac{9}{32}$	$\frac{1}{4}$
	33		$\frac{15}{16}$	$\frac{7}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$8\frac{1}{4}$	$9\frac{3}{16}$	$18\frac{9}{16}$	$18\frac{13}{16}$	$7\frac{7}{16}$	$7\frac{3}{16}$	$\frac{9}{32}$	$\frac{1}{4}$
	25	Quarter Stroke	$1\frac{3}{8}$	$1\frac{1}{4}$	$\frac{5}{16}$	$\frac{5}{16}$	$6\frac{3}{16}$	7	$16\frac{7}{8}$	$17\frac{1}{4}$	$9\frac{1}{8}$	$8\frac{3}{4}$	$\frac{9}{32}$	$\frac{9}{32}$
	20													

It is claimed by the builders of this valve gear, that excessive compression, and back pressure in the short cut-off is entirely eliminated.

No change is required in the valves or auxiliary parts to the main valve as at present in use, merely a change in the imparting motion accomplishes the improvements set forth.

No change of construction or different parts are required for either inside, or outside admission, except the position of the eccentric crank, which leads the main crank for outside admission and follows it for inside admission. It is also claimed by the makers (The Pilliod Co. of Swanton, O.), that owing to the low terminal pressures, a larger exhaust nozzle may be used. Tables 48 and 49 give comparisons of valve action with the Baker-Pilliod valve gear, and with the Stephenson link.

The following claims for efficiency are made by the manufacturers of the Baker-Pilliod valve gear:

1st. The weight of the gear as compared with the Stephenson Link Gear represents about 40% of the latter's weight; about 60% of the weight of the Walschaert.

2d. It is a universal application for all types of engines. Absolutely outside of frame, and all power reciprocating parts, with perfect accessibility to all parts. It requires no special design for the different style, type or class of locomotive, reducing first cost and facilitating repairs at minimum costs.

3d. The imparting motion is of a very low throw of 6½" and speed has no destructive or high deteriorating effects. Its reciprocating parts are as light as is consistent with the labor to be performed. No loose, sliding or lifting joints or link blocks. Each and every connection of the gear is mechanically positive.

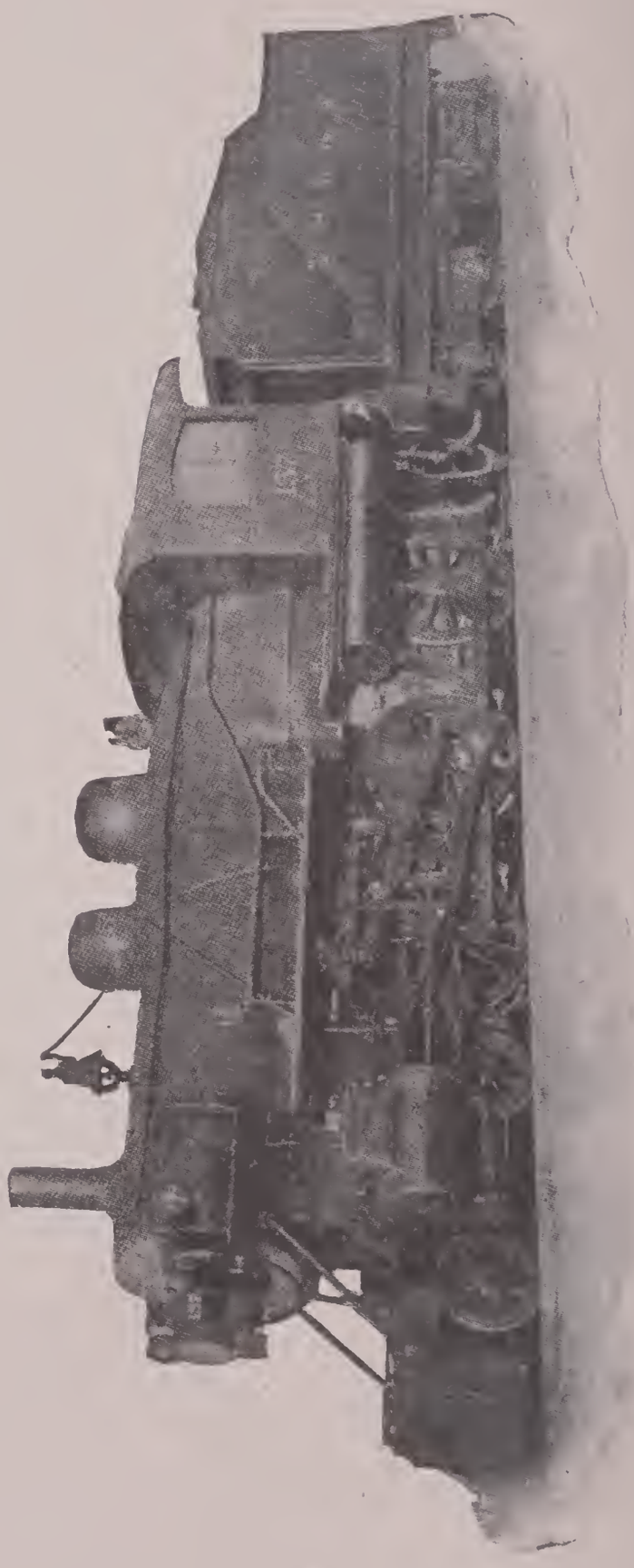


Fig. 27. Engine equipped with Baker-Pilliod Valve Gear.

4th. The reverse requires no change in the reciprocating parts, no readjustments of its bearings or alignment, merely a movement of its positive connected radius arm, overcoming all objections of any type of valve gear that employs a link movement.

5th. It produces a greater range of valve events than any other type of valve gear, for the reason that its various positions are a positive and direct connected movement.

6th. It is the only gear employing a single valve that produces dwell upon the opening and closing of the valve, resulting in an extremely quick opening and quick closing valve, requiring only a travel of about 5% of the piston to have full port opening.

7th. It maintains uniform lead at all points of cut-off.

8th. The movement of the valve produces the highest effective pressure upon the piston. Five per cent of the piston travel required for full port opening; a dwell upon the opening of 40% of the piston travel; 25% of the piston travel to close; and 25% of the travel effective expansion, with full travel of the valve.

9th. It produces 25% higher "Range of Temperatures" than any valve gear employing a single valve from initial to effective pressure upon the piston.

10th. It is especially adapted to high speed, producing efficiency equal to valve gears employing 2 to 4 valves.

11th. We guarantee to produce a saving of 5% in fuel, uniform lead, uniform cut-off, late release, late compression, balanced compression, total absence of preadmission, 5% increase actual traction power transmitted to wheels.

Table 50 gives weights of parts.

Table 50.—Weight of Baker-Pilliod Valve Gear.

	Material.	Weight.	Number of parts required.	Total weight.
Eccentric rod	Forged Steel	135	2, used R & L	270
Eccentric arm	C. S.	84	1-right, 1-left	168
Eccentric crank	C. S.	73	2, used R & L	146
Combination lever	Forged Steel	81	1-right, 1-left	162
Crosshead yoke	C. S.	61	1-right, 1-left	122
Union link	C. S.	27	2, used R & L	54
Combination lifter	C. S.	31	2, used R & L	62
Valve rod	C. S.	27	2, used R & L	54
Bell crank	C. S.	80	1-right, 1-left	160
Reverse yoke	C. S.	115	2, alt. for R & L	230
Radius arm	C. S.	49	2, 1 used R & L	98
Frame inside half	C. S.	165	1-used, right	165
Frame inside half	C. S.	165	1-used, left	165
Frame outside half	C. S.	180	1-used, right	180
Frame outside half	C. S.	180	1, used left	180
Reverse arm	Forged Steel	55	1, used right	55
Reverse arm	Forged Steel	27	1, used left	27
Reverse shaft	Forged Steel	130	1-used	130
Reverse shaft box	C. S.	80	2, (1 used R & L)	160
Frame bracket	C. S.	66	2-(1-R, 1-L)	132
Frame tie plate.	Bar Iron	200	2-used	400
Guard & frame c'n't'n	C. S.	73	2-(1R, 1-L)	146
			TOTAL WEIGHT	3236

Weight of Moving Parts, each side 514 lbs.
Total Weight of Moving Parts, 1028 lbs.

JOY'S VALVE GEAR.

Of all radial gears this has been most extensively applied to locomotives, and it is probably the best radial gear known so far. It comprises few parts and gives—theoretically at least—a faultless distribution. In this gear (Fig. 28) motion is derived indirectly from a point B in the connecting-rod through the interposition of rods B C and C D. The end of the rod, or floating lever, E G, which is attached to the valve-rod, in one-half revolution of the crank from dead center to dead center, travels through a distance equal to twice the lap and lead, in addition to which, owing to the vertical movements of the points B and E, its center, G, moves along the curved slide so as to determine the port opening. Reversal is effected by moving the curved slide. Since this movement does not affect the first-named component of the motion of the lever E G, it follows that the leads are fixed.

Hence the problem to be solved in designing this gear is to choose such a position of B that (1) the travel of the end of the lever E G will equal twice the lap and lead, and (2) that the vertical vibration of the point G along the slide will have the effect of imparting to the valve sufficient movement to give the desired port opening.

When G has reached its highest point in the slide, and commences to return, both G and F act together to produce a quick cut-off. The rise and fall of the axle relatively to the engine frame under the action of the springs affects the distribution with this gear to some extent.

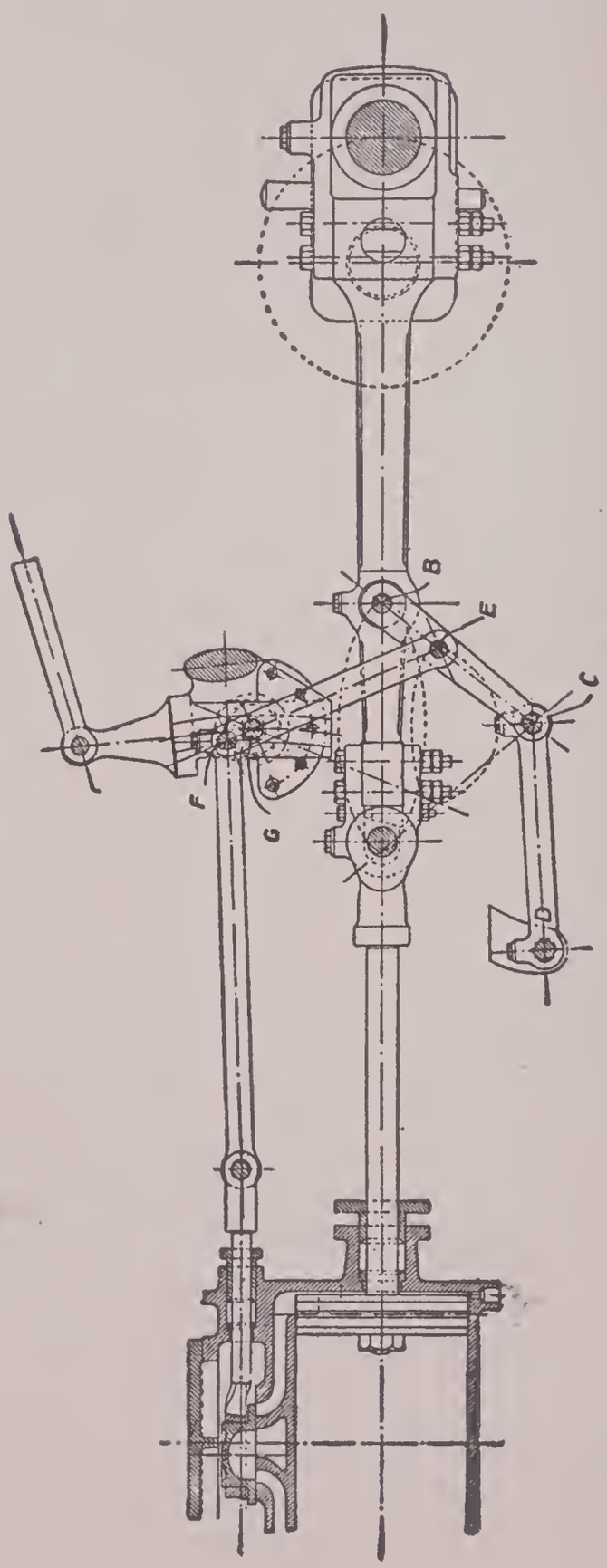


Fig. 28. Joy's Valve Gear.

The same may be said as to inevitable wear at the joints, which are more numerous in gear of this kind than in the case of the link motion.

The connecting rod has an enlarged boss formed in it at a suitable point, about one-third of the length of the rod measuring from the small end. This is bored out and fitted with a bush; through it a pin passes, projecting on either side to carry the forked ends of the "correcting link." This is coupled at its other end to the "anchor" link, which in turn is pinned and allowed to vibrate about a fixed point. In the illustration this point is upon a projecting lug upon the motion plate or slide bar bracket. Examples may, however, be often met with in which it is on either a frame stay or a shaft fixed across the frames, as found most suitable in the particular design of engine. The correcting link has a bearing in its central portion, to which is connected the "valve levers," these being further provided with two other bearings—one at the top end for the attachment of the valve rod end and the other close to it for the pin, which carries upon its ends the "quadrant blocks," which are fitted to sliding curved guides fixed to the reversing shaft.

It will be seen that the vibration of the connecting rod when the engine is running moves the quadrant blocks up and down in the curved guides, which compel it to take a course dependent upon the position of the guides; thus, in the sketch the guides are vertical, therefore the valve rod, and with it the valve, will have least movement horizontally, when the engine is in "mid-gear." When it is desired to run forwards, the reversing shaft has a partial revolution given to it, tilting the top of the guides over towards the cylinders; this causes the block

to move in the required position and gear. When the engine is reversed to full back gear, the shaft is partially turned, so that the top of the guides lay over towards the fire box; the points of cut-off between mid-gear and full gear are settled by giving the guides more or less inclination as required.

QUESTIONS.

192. From what two sources is the motion of the valve derived in the Baker-Pilliod valve gear?

193. How is the point of cut-off controlled?

194. What actuates the lap and lead?

195. In mid-gear with reverse lever in the center notch, what motion is imparted to the valve?

196. What is the effect of moving the reverse lever forward from center notch?

197. What is necessary in order to cause the engine to run backward?

198. Mention some of the advantages claimed for the Baker-Pilliod valve gear.

199. How does it compare in weight with the Stephenson and Walschaert valve gear?

200. From what source does Joy's valve gear receive its motion?

201. Is this motion direct?

202. Is there a constant lead at all points of cut-off with the Joy valve gear?

203. Does the vertical movement of the axle relative to the frame affect the steam distribution with the Joy valve gear?

204. What may be said regarding the wear at the joints in a radial valve gear?

205. When it is desired to run forward what changes take place in the positions of the various parts of the Joy valve gear?

206. Describe the positions of these parts when engine is reversed.

TYPES OF LOCOMOTIVE VALVES.

The principal objection to the use of the D slide valve is the large amount of friction caused by the action of the steam pressing the valve against its seat, and inventors have racked their brains for many years in efforts to produce a valve that would work without friction, and at the same time give a correct distribution of the steam to and from the cylinders.

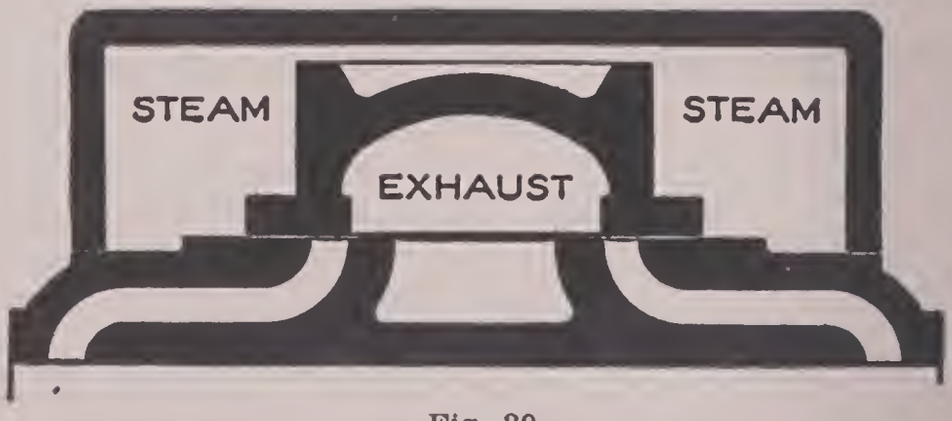


Fig. 29.

The piston valve, while practically balanced, owing to the pressure of the steam acting upon each end, is, nevertheless, not a perfectly balanced valve unless the valve rod extends through both ends of the valve chamber, and this necessitates an extra gland and set of rod packing. In order to more clearly illustrate this idea, reference is made to Figs. 29 and 30. Fig. 29 shows a plain D slide valve, and it will be noticed that the full pressure of steam in the valve chest acts upon the back of the valve. Of course there is a certain amount of back pressure from the steam port and exhaust port that tends to over-

come the direct pressure; still there is an enormous strain on the valve gear that is required to move a valve under such conditions. Fig. 30 shows a solid piston valve with outside admission, being thus identical in action with the D valve.

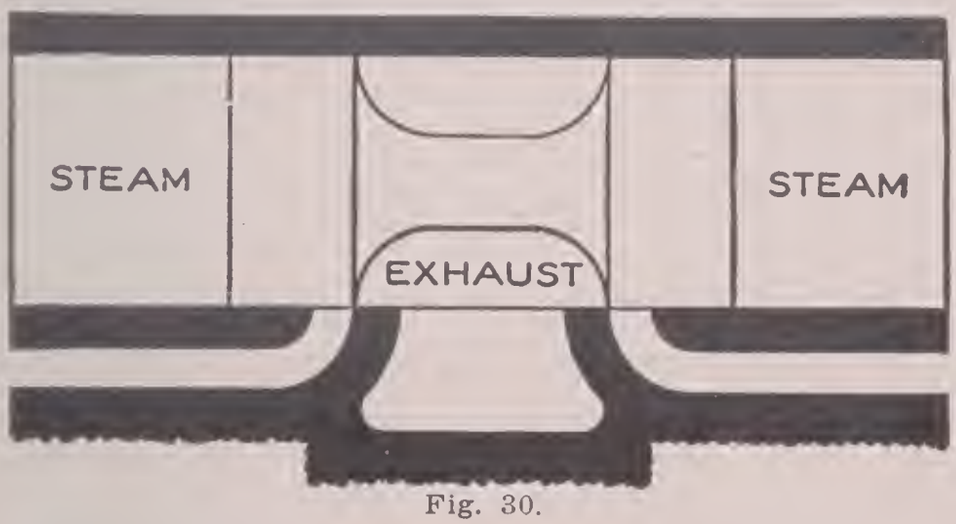


Fig. 30.

No valve rod is shown in either cut, but it will easily be seen that with the valve rod attached to but one end of the piston valve the area of that end will be decreased

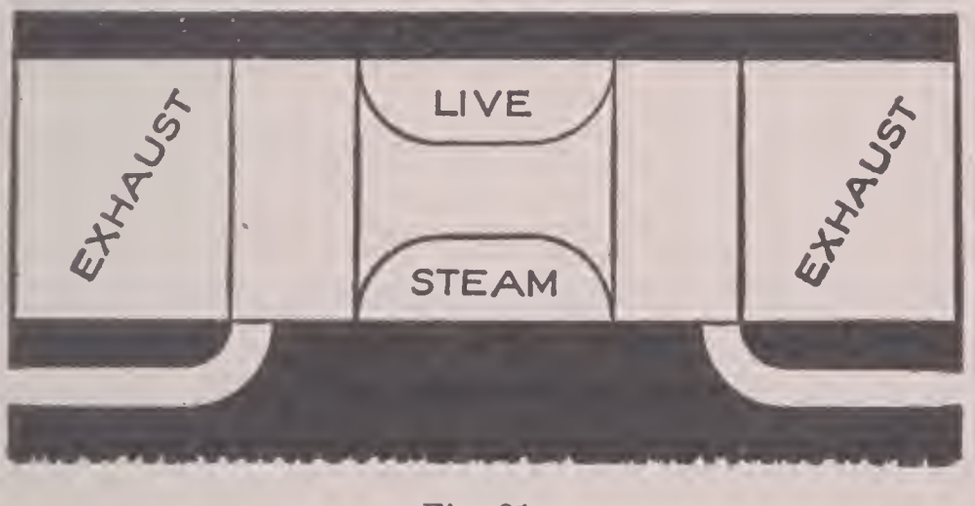


Fig. 31.

just so much, and the valve will be unbalanced by an amount equal to the sectional area of the valve rod, but this amount is so insignificant that builders very seldom

add the extended valve rod, and so the piston valve may be considered as balanced, the only friction being that due to the weight of the valve and the friction of the packing rings when the valve is fitted with them. In some types of piston valves the live steam is admitted inside, between the heads, as shown in Fig. 31, and the exhaust passes out around the ends, but the same principle of balancing is retained as with the outside admission type, for the reason that the pressure is applied between the ends of the valve instead of on the outside as with the other type. The sketches here given do not show the valves in their true proportions, being merely used to illustrate the principle upon which the piston valve works. In practice the valve is made as long as possible, in order that the ports leading to the cylinder may be shortened to the minimum.

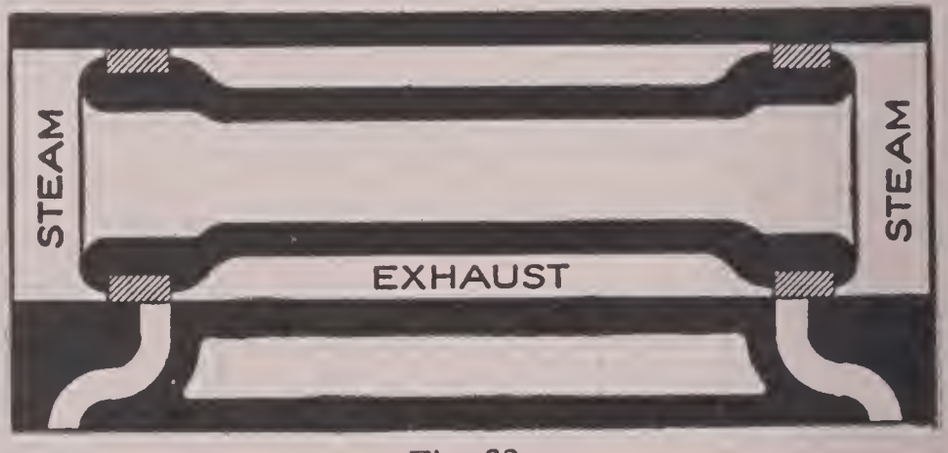


Fig. 32.

Another type of piston valve is shown in Fig. 32. This valve is made hollow for lightness and has packing rings at each end to prevent the steam from passing into the ports until at the proper moment. The edges of these packing rings control the admission of steam to the ports in the same manner as do the edges of the D valve, and when the valve is one of outside admission it is set in

the same manner as the D valve is. But if admission is from the inside, as shown in Fig. 33, the movement of the valve is reversed, as is the method of setting also. As it is very essential that the packing rings at each end of a piston valve be steam-tight, a certain element of friction is introduced in this manner. In the larger number of cases where piston valves are used, central or inside admission is the rule, a great advantage of this type over outside admission valves being that the larger portion of the cooling surface of the valve chamber is reserved for the exhaust steam. Another advantage is that of having only exhaust pressure against which to pack the valve rods, and make the joints for the heads of the valve chamber.

Fig. 33.

In taking charge of an engine having piston valves, an engineer should always first "look her over" and note the positions of the eccentric with relation to the crank pin. He should also take a look at the rocker shaft if there is one. He will then be able to satisfy himself as to whether the valves have outside or inside admission, a very important thing to know in case anything should happen out on the road that necessitated resetting of one or both of the valves to enable him to bring his engine

home. As before stated, the movement of a piston valve having outside admission is precisely the same as that of a D slide valve, but it is well to note the fact that while the great majority of engines fitted with D slide valves have indirect valve gear, still there are some in which the motion is direct. For the guidance of the engineer in such cases, the following four simple rules are here given:

Rule 1. If the eccentric and crank pin are together, that is, on the same side of the driving shaft, and there is a rocker arm that reverses the motion, the valve has outside admission, indirect.

Rule 2. If the eccentric and crank pin are together and there is no rocker arm, but direct motion, the valve has inside admission, direct.

Rule 3. If the eccentric and crank pin are on opposite sides of the driving shaft, and there is a rocker arm to reverse the motion, the valve has inside admission, indirect.

Rule 4. If the eccentric and crank pin are on opposite sides of the shaft, and there is no rocker arm to reverse the motion, the valve has outside admission, direct.

The American Balanced Valve Company of Jersey Shore, Pa., are the makers of a new type of piston valve, which they term "The American Semi-Plug Piston Valve." This valve, a description of which is here given, has performed very efficient service since its introduction, and it appears destined to occupy a prominent position in locomotive work in the future.

Referring to Fig. 34, an internal admission valve is shown. The inner sides of the two snap rings, 1-1, are beveled. The outer sides of the snap rings are straight and fit against the straight walls of the valve spool.

Against the beveled sides of the snap rings, solid, uncut, non-expansible wall rings, 2-2, fit. Their inner sides are beveled at a greater degree of angle than their outer sides, which fit the snap ring.

In between the two wall rings is placed a central double tapered snap ring, 3. This ring is properly lapped, and is put in under tension, thus holding the wall rings apart. putting a slight grip on the snap rings laterally. Thus applied, the action is as follows: When steam is admitted to the steam chest, or central portion of the valve, it

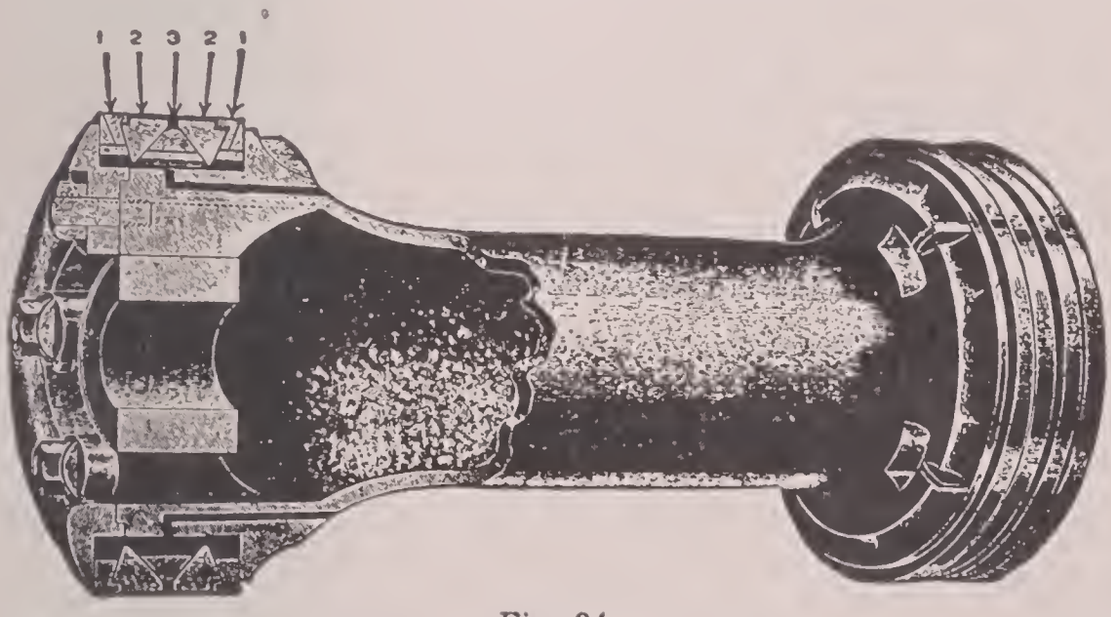


Fig. 34.

passes through openings in the spool to the space beneath all of the rings, and acts upon the central wedge ring direct, giving it a lead of the snap rings in action, and forcing the wall rings against the sides of the snap ring, so that prevention of their excessive expansion is positive. The snap rings are thus expanded against the casing just enough to make steam-tight contact, and the central ring grips them there, and they are prevented from further expansion. This is demonstrated by withdrawing the valve from the valve chamber while under

steam until the first ring in the spool is entirely out of the cylinder, when no increase in the diameter of the snap ring can be observed. It can then be pushed back into the cylinder again. It will readily be understood how easy it is to prevent further expansion of the snap ring by the pressure underneath it, when the degree of angle of the bevel on the inside of the snap ring is considered. By making this degree greater, the power of the central wedge ring would be sufficient to decrease the diameter of the snap ring, closing it away from the valve chamber. Therefore it appears that this valve has all the advantages of the plug valve, without the drawbacks of the plug valve, and it has all the advantages of the snap ring valve, without the drawbacks of the snap ring valve, because it is practically a plug that does expand and take care of itself, not only for the difference in contraction and expansion, but also for wear; yet the plug is not so rigid as to knock a cylinder head out before relieving the water from the cylinder, and yet it is absolutely adjusted to the diameter of the casing at all times, and is held there and allowed to get no larger during its work under pressure. The rings are so lapped that they are steam-tight from all directions, and the bevel lap joint maintains unbroken steam and exhaust lines at the edge of the ring.

Fig. 35 further illustrates the construction of this valve, giving end and sectional views of the different parts. A common defect of snap ring piston valves is that the steam pressure gets under the rings, and expands them against the casing with the full force of the chest pressure, thus causing excessive friction, while at the same time the cage is worn unevenly by the valve working at short cut-off and over the ports. Under such conditions,

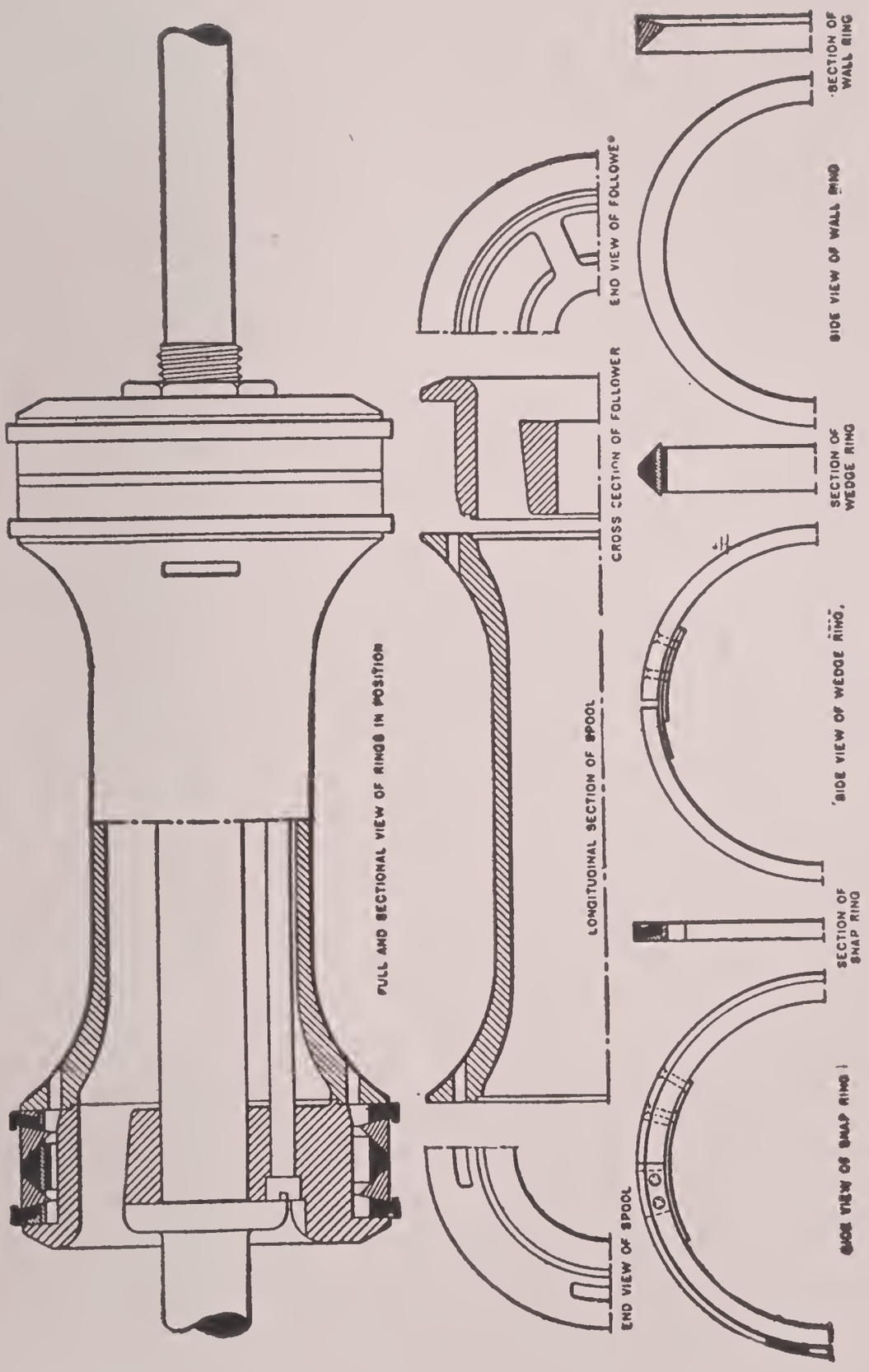

FULL AND SECTIONAL VIEW OF RINGS IN POSITION
LONGITUDINAL SECTION OF SPOOL
CROSS SECTION OF FOLLOWER
END VIEW OF FOLLOWER
END VIEW OF SPOOL
SIDE VIEW OF SNAP RING
SECTION OF SNAP RING
SIDE VIEW OF WEDGE RING
SECTION OF WEDGE RING
SIDE VIEW OF WALL RING
SECTION OF WALL RING

Fig. 35.

steam-tight joints soon become leaky, and the leakage rapidly increases as the wearing goes on.

A piston valve, in order to give efficient steam-tight and durable service, should automatically regulate the frictional contact of the rings against the cage, and keep the cage perfectly true.

Many locomotives are equipped with piston valves of different types, but the internal admission valve appears to be the favorite.

One of the advantages the piston valve possesses over other forms of slide valves is that it may be made long enough to bring the two faces, or working edges near the ends of the cylinder, thus greatly reducing the clearance between the valve face and the piston.

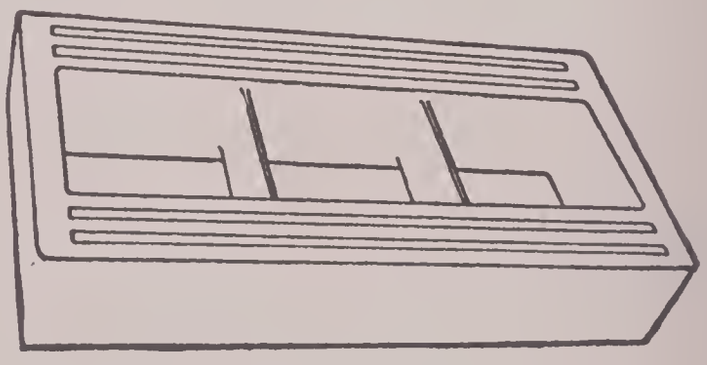


Fig. 36.

The term balanced valve, as used in this connection with reference to locomotive practice, is meant to include all balanced valves except those of the piston type. As stated at the beginning of this section, there have been many different kinds of balanced valves applied experimentally to the locomotive, by inventors in their efforts to reduce the friction between the face of the valve and its seat.

A few of the more meritorious of these will be described and illustrated. Fig. 36 shows the Jack Wilson High Pressure Valve.

This valve has two faces. One face moves on the valve seat proper, and the other face moves in contact with the face of the balance plate. Both faces are the same, and it has no crown, but is open throughout. The face of the balance plate is an exact duplicate of the cylinder valve seat, and set in alignment therewith, therefore whatever conditions exist on one face of the valve must necessarily exist on the other face.

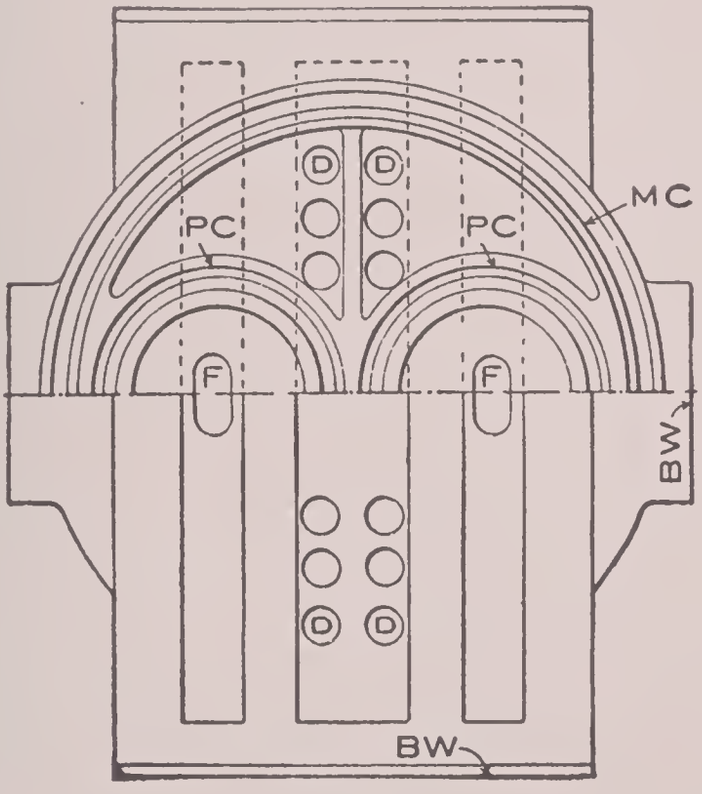


Fig. 37.

The walls of the valve are provided with ports which pass from face to face of the valve. The balance plate, Fig. 37, contains balancing cones M C and P C (main cone and port cone), and two centering ring grooves, that register with similar grooves in pressure plate.

The face of the balance plate, Fig. 38, is an exact duplicate of the cylinder valve seat and forms a second valve seat against which the valve operates in unison with its operation on the cylinder valve seat, the second seat being held by means of the centering rings CR, Fig. 39, in exact alignment with the valve seat proper. The back or opposite side of the balance plate, Fig. 42, contains the following cones: one large or main cone (MC) and two small or port cones (PC) on the interior of the main

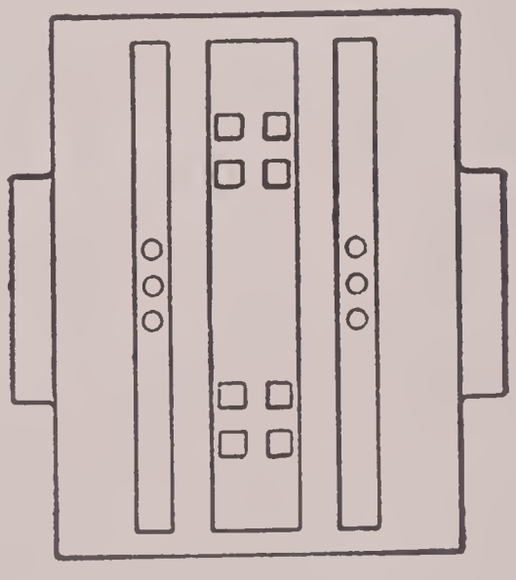


Fig. 38.

cone, and on which the packing rings are placed, which forms the balancing feature to the valve, and the centering ring cones. The balance plate is provided with wings (BW), which fit 1-16 inch loose into the wings of the pressure plate (or into the steam chest itself), preventing excessive movement of plate. Taper or beveled packing rings set on the cones form joints against the pressure plate.

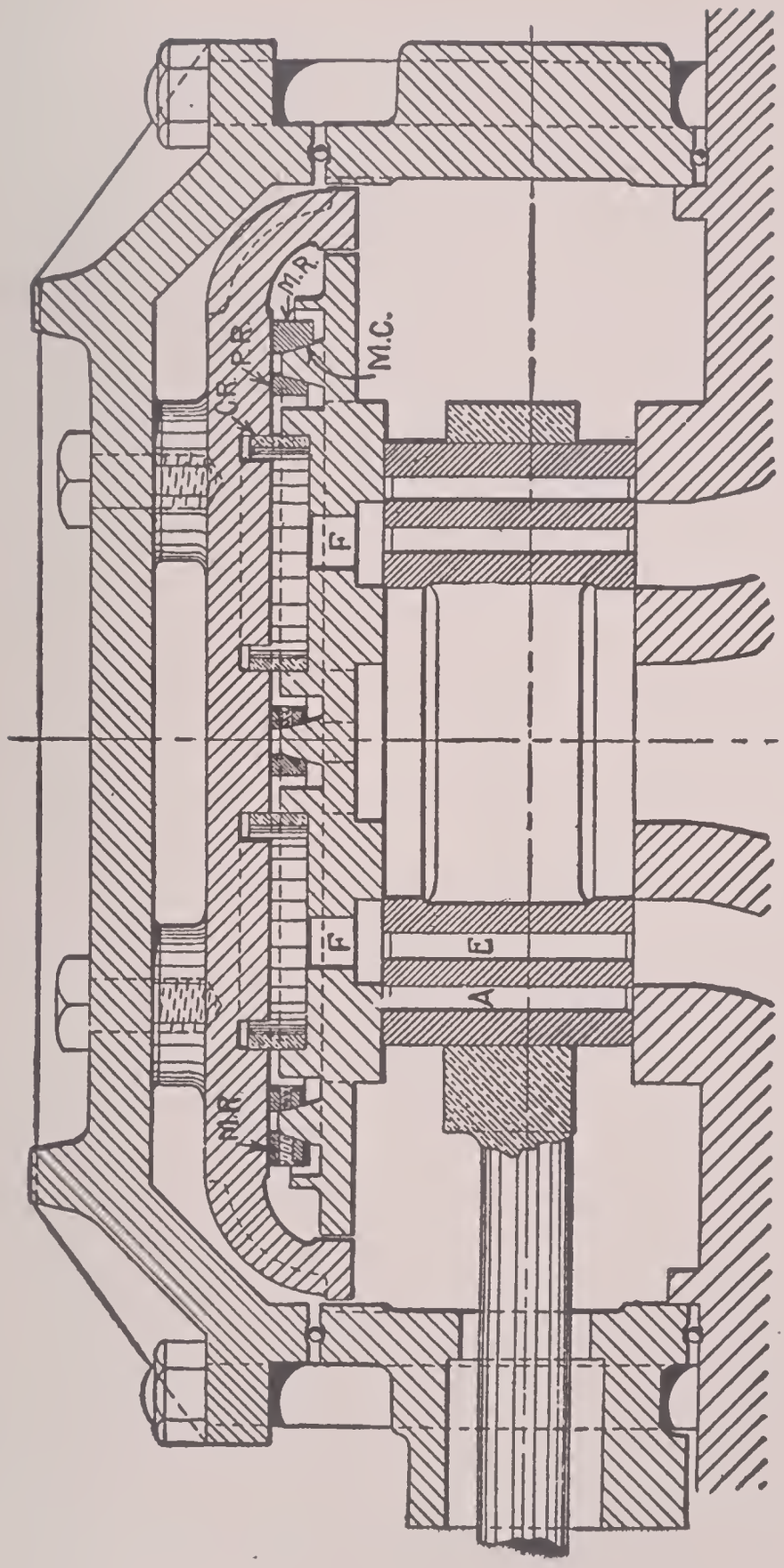


Fig. 39.

Pressure Plate. The pressure plate is made either as a part of, or separate from the chest cover. In the type of valve here referred to, the pressure plate is made separate (see Figs. 40, 41 and 42), and is provided with wings (W) which are machined to fit snugly into the steam chest; the chest being first centered with the valve seat by fitting over lugs on the cylinder, or by dowel pin,

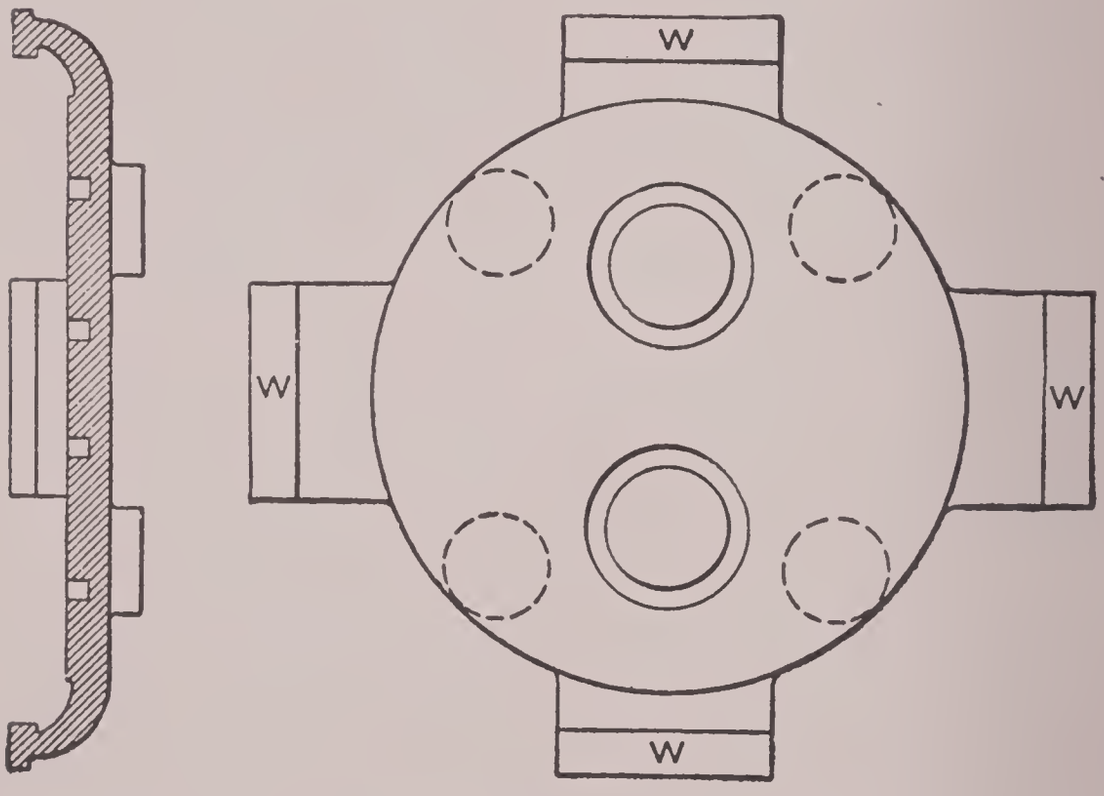


Fig. 40.

and machined at the top to receive wings (W) of the pressure plate. Into the face of the pressure plate two grooves are cut with either straight or taper walls and which register correctly with the corresponding grooves in the balance plate; these are called centering ring grooves and into them two centering rings (CR) are placed, slightly under tension. Under normal conditions these steel rings hold the balance plate in alignment with

the valve seat, but under abnormal conditions, such as dry valves, the strain will be taken by the wing of the balance plate, against the wing of the pressure plate, preventing excessive contraction of the centering rings. Against the face of the pressure plate the balancing rings

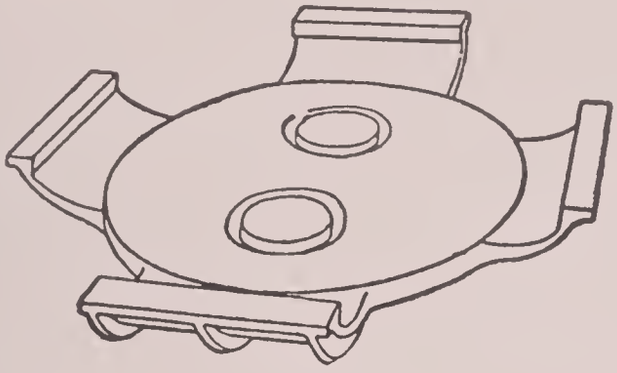


Fig. 41.

form steam joints. The balanced area of this valve is changeable, the change taking place automatically, so as to correspond with the changed condition of the valve on its seat at different points of its travel.

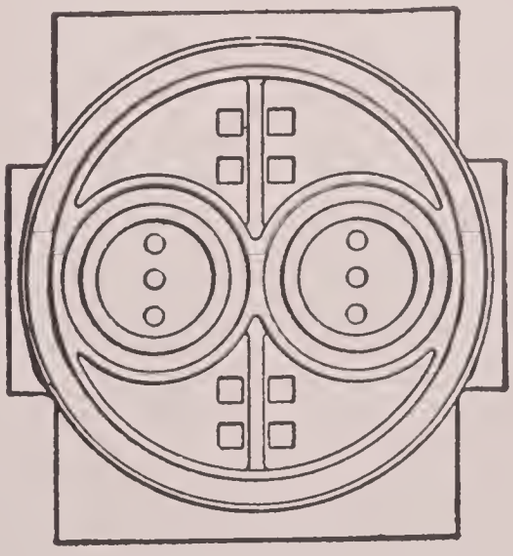


Fig. 42.

Referring to assembled cross sectional view, Fig. 39, the valve is seen in central position on the seat and the upper seat or face of balance plate in position correspond-

ing with the valve seat. The steam chest is centered by machined faces fitting over machined lugs on the cylinder: on old power, dowel pins are used. The chest has finished strips at top into which the finished ends of the wings of the pressure plate (W) fit snugly, thus insuring the central position of the pressure plate over the valve seat. The finished wings of the balance plate (BW) fit 1-16 inch loose between inside faces of the wings of the pressure plate, but the balance plate is held perfectly central by two steel centering rings (CR). The tops of the cones on the balance plate are 1-8 inch from the face of the pressure plate, allowing the balance plate to lift 1-8 inch off from the valve, which affords perfect relief to the cylinder while the engine is drifting and for the relief of water from the cylinder. This 1-8 inch clearance in height adjustment must be maintained. The main balancing ring (MR) is made the proper diameter to balance the valve as great as possible while in its central (or heaviest) position, there being just sufficient area left to insure the balance plate being held steam-tight on the valve. The interior of the main ring is open to the atmosphere through the holes D, which lead to the exhaust cavity of the valve.

The valve is thus balanced so that it will move perfectly easy in its heaviest position, but conditions are changed by the opening of a steam port (and at instant of cut-off. See Fig. 43), at which time the ordinary slide valve is subjected to the upward pressure of the steam in the cylinder port, and if properly balanced in central position would, at this position, be thrown off its seat, but in this valve the port pressure (whatever it may be) has free access to both sides of the valve by reason of the passages through the valve to the port in the face

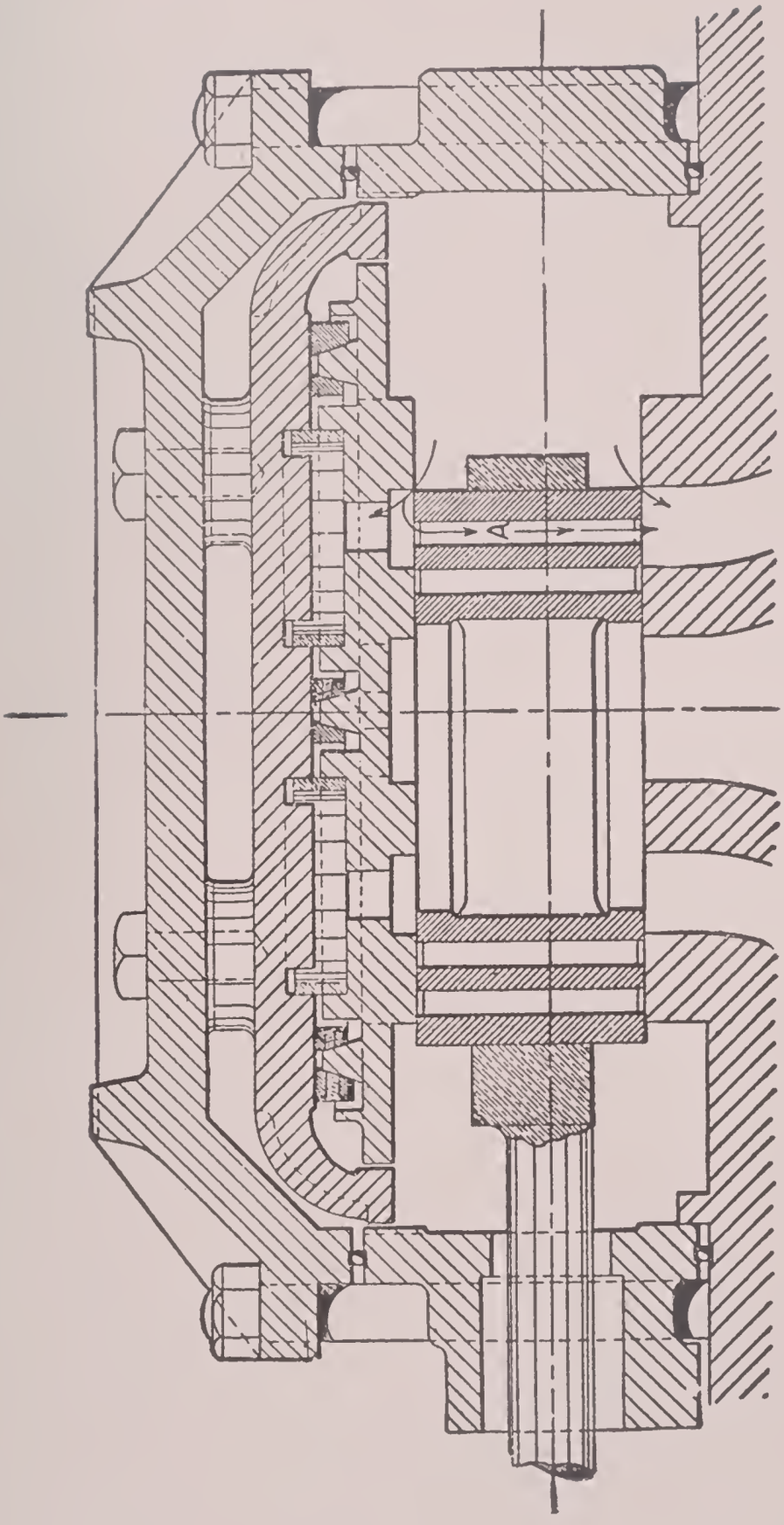


Fig. 43.

of the balance plate which corresponds with the cylinder port: therefore the pressure in the port has no effect whatever upon the valve, it being on both sides of the valve face of equal area, and pressure is, therefore, equalized so far as the valve is concerned, but the pressure in the port of the balance plate would lift the plate off from its seat on the valve if it was not so equalized, or annulled; therefore a port ring (PR), of proper area to balance this pressure, is placed over each port in the inside of the main ring on the top of the balance plate and is open to the port through passage F, Fig. 39, so that a pressure equal to that in the steam port is always on both sides of the balance plate, as well as on both sides of the valve, and the port pressure is rendered inoperative on the valve or on the balance plate. Communication from the cylinder port, through the valve and through the balance plate to the interior of the port ring PR, cannot be shut off at any time, but is maintained throughout the travel of the valve. Therefore the same pressure that is in the port at any given time is also on both sides of valve and pressure plate in the same area, and the port pressure is, therefore, not considered in figuring the main balance for the valve.

There is another position of the valve during its stroke where the slide valve is subjected to an upward pressure, or pressure against its face, which tends to lift it from its seat; that position is at over-travel of the valve face over the valve seat; this position is shown in Fig. 44, but in this valve it will be observed that the top face or back of the valve travels out from under the seat of the balance plate exactly to the same extent that it over-travels the cylinder seat, and pressure is, therefore, equal on both sides of that portion of the valve that is over the seat at

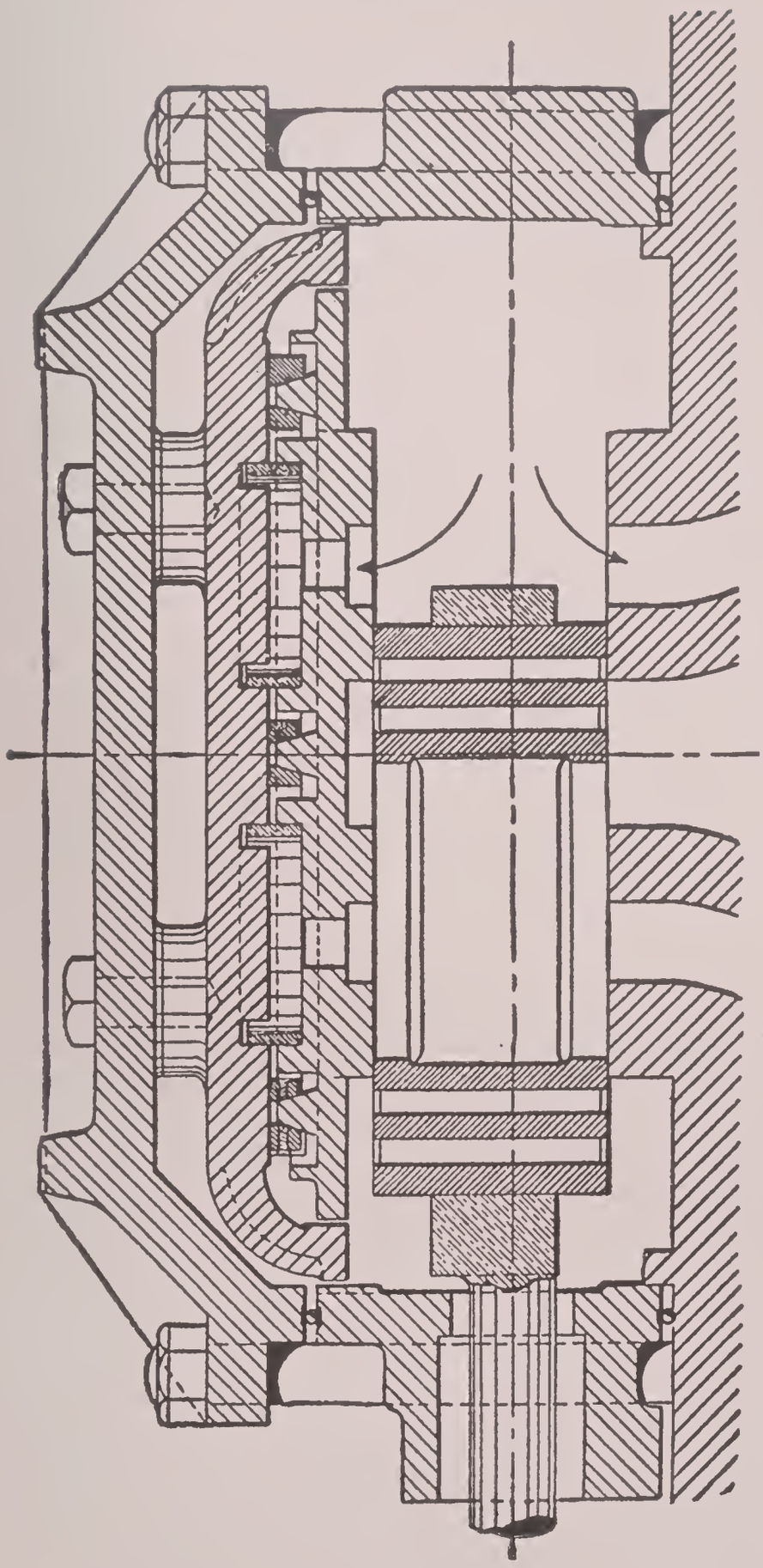


Fig. 44.

any point of travel. With the main ring balancing the valve fully in its central or heaviest position, the port ring balancing the port pressure, and over-travel of the valve on its seat being equalized by equal pressure at top and bottom, it will be clear that the valve is fully balanced in all positions of stroke, and is, therefore, available for high pressures.

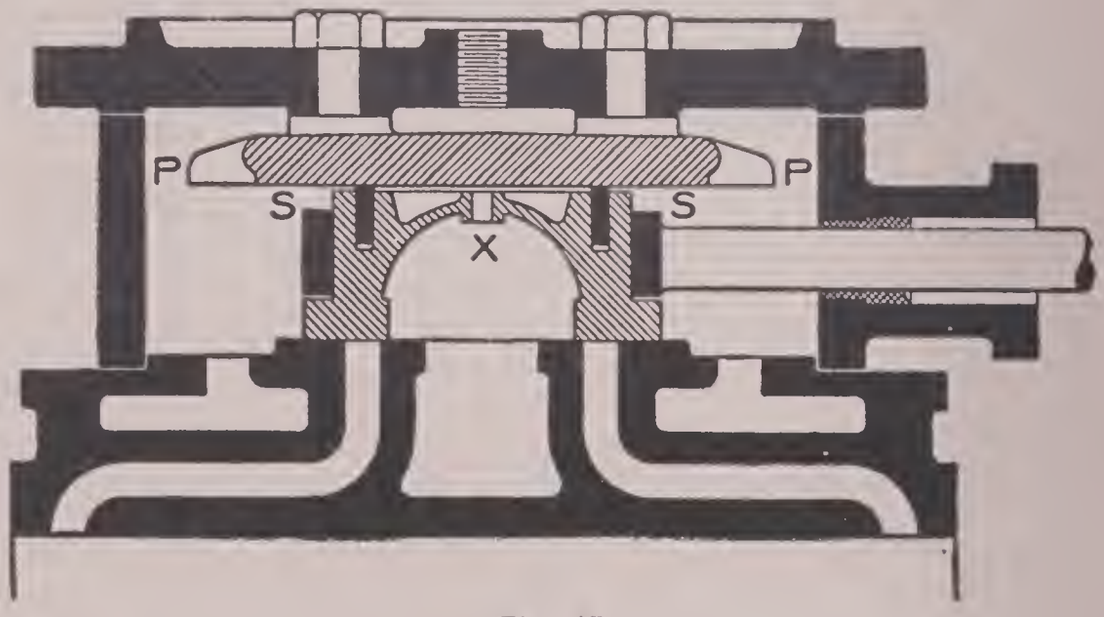


Fig. 45.

The Richardson Balanced Valve. This form of balanced slide valve, together with the Allen-Richardson balanced slide valve, is manufactured by H. G. Hammett of Troy, New York, and is used largely on locomotives. Figs. 45 and 46 represent transverse and longitudinal sections through the center of an ordinary locomotive steam chest fitted with the Richardson valve. Fig. 47 shows a plan of the valve, and Fig. 48 is an elevation of one end of the packing strips and spring, the only alteration being the addition of the balance plate, PP, Fig. 45, and the substitution of a valve adapted to receive the packing strips S, S, S, S.

It will be noticed in this instance that the balance plate is bolted to the cover of the steam chest, but these may be cast in a single piece. The four sections of packing

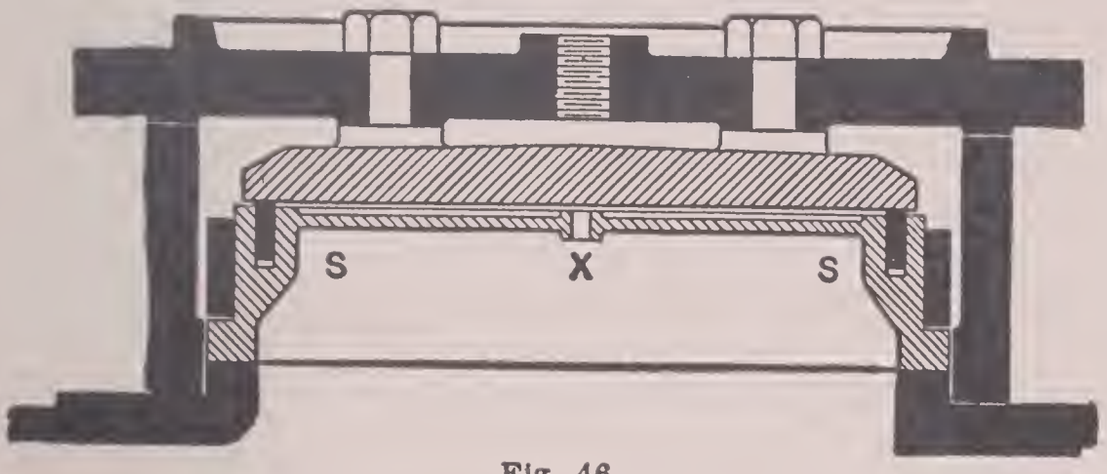


Fig. 46.

enclose a rectangular space, R, Fig. 47, which equals in its area the total amount of valve surface which is to be relieved of excess pressure, the packing strips preventing

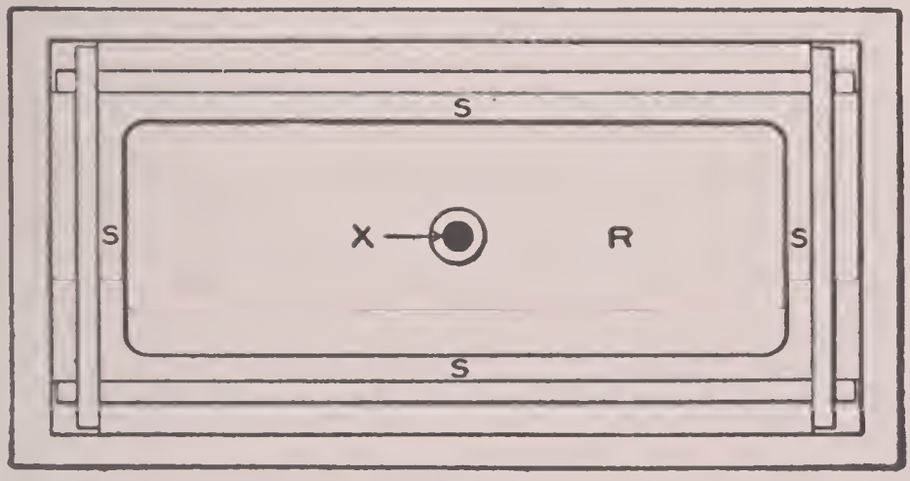


Fig. 47.

the steam from entering this space, and the small hole, X, communicating with the exhaust cavity in the valve, relieves space, R, from any possible accumulation of pressure.

The four packing strips consist of two longer ones, which are simply rectangular pieces of cast iron, while the two shorter ones, Fig. 48, have gib-shaped ends to retain them in their proper position. Beneath each packing strip a light elliptic spring, shown in Fig. 48, is placed, which holds these strips in position against the balance plate when the steam is shut off.

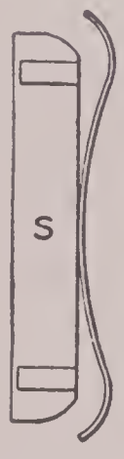


FIG. 48.

In operation these different sections maintain a steam-tight contact, by a direct steam pressure, with the balance plate and with the inner surfaces of the grooves provided to receive them, the joint being secured by the abutting of the ends of the two longer sections against the inner surfaces of the gibbed sections at the four corners.

THE ALLEN-RICHARDSON BALANCED SLIDE VALVE. The Allen valve is designed to at least partly prevent the wide-drawing of the steam, when high speeds are maintained with the valve cutting off early in the stroke.

In the Allen valve, an additional passage for the inlet of steam is furnished, as will be clearly seen by referring to Figs. 49 and 50. These are transverse and longitudinal sections through the valve and steam chest, and it will be noticed that, when the steam port is open one-half inch in the ordinary manner, the port of the cored passage is also open to a like extent on the other side of the valve; consequently the effective area of the steam port is doubled, and is thus the actual equivalent of a single port with a one-inch opening.

The wire-drawing incident to running at high speeds with the valve cutting off early in the stroke is thus greatly diminished, with a resultant economy of steam and fuel. A reduction of wire-drawing carries with it

a higher average pressure on the piston when working at a similar cut-off; consequently the usual average pressure can be maintained with a shorter cut-off, resulting in an appreciable economy. While the unbalanced Allen

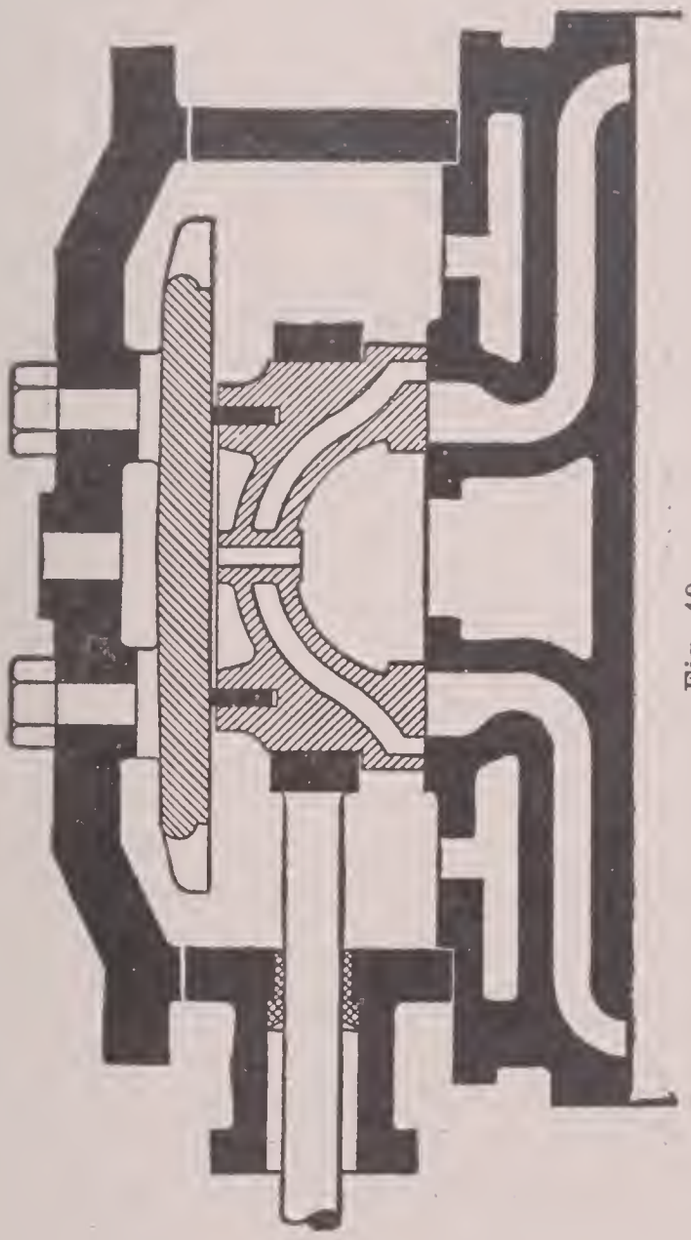


Fig. 49.

valve, therefore, secures a better and more economical distribution of steam, its use entails certain disadvantages.

On the face of a slide valve, the area of bearing surface is never sufficient to secure its wearing well under

a heavy steam pressure; and this wearing surface is yet further reduced in the Allen valve, owing to its internal steam ports. This internal passage actually divides the valve into two parts, and the steam pressure, acting on the outer part, springs and bends its working face below that of the internal or exhaust port of the valve. The available wearing face is consequently reduced to a space about one-half as wide as the outside lap of the valve, and this fully accounts for the rapid wearing of the un-

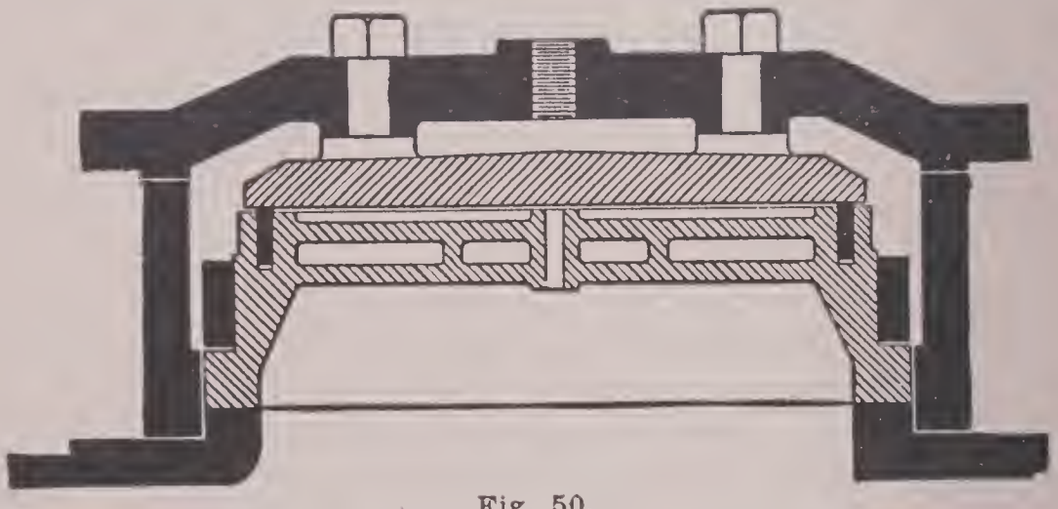


Fig. 50.

balanced Allen valve, and for the trouble and expense of constantly refacing valves and seats, and the loss of the steam blown through leaky valves quite offsets the advantages gained by a reduction of wire-drawing.

These manifest disadvantages are entirely overcome by a proper balancing of the valve, which secures all of the advantages of the Richardson device, plus an increased steam economy resulting from using the Allen ports.

In order to secure the best possible results from the employment of the Allen unbalanced valve, its ports and bridges should exceed the full travel of the valve, by at least one-eighth of an inch, and the radius of the link should always be as long as permissible to escape an

excessive increase of lead when cutting off early in the stroke. This latter, however, is prevented with the Walschaert valve gear, which gives a constant lead at all points of cut-off.

The Allen valve has made fair progress in American locomotive practice, but there is still considerable prejudice against it. This prejudice, we believe, is founded in want of experience, or in a mistaken view of what the capabilities of a valve are. While this conflict of opinion continues to exist, we think that some remarks made by Mr. E. M. Herr, superintendent of motive power of the Northern Pacific Railway, at a meeting of the Western Railway Club, might be studied to advantage. The discussion was on lead in the setting of locomotive valves, and in this connection he said:

"With a very long port you can give an engine less lead than with a short port, and the kind of valve used has also an effect. With an Allen ported valve you can still further reduce the lead and get the same work out of the engine. The Allen valve, in my opinion, is a very valuable device if rightly used, and I believe that many railroad men condemn it because they have not used it in just the right manner. You do not get the full advantages of an Allen valve if you give it anything like as much lead as you would a plain valve. One of the principal advantages of an Allen valve is that you can reduce the lead and still retain the mean effective pressure in the cylinder. Of course, there is another advantage with the Allen valve, and that is the more rapid admission of steam into the cylinder, and this enables a locomotive at high speeds, with an Allen valve, to very much exceed in power the same engine with a plain valve.

"In making some tests on the North-Western testing plant not very long ago, we showed very conclusively that at high speeds a 16x24-inch engine would develop more power with an Allen valve than a 17x24-inch engine, with practically the same size of driver, would develop with a plain valve. In fact, a 16x24-inch engine on a certain division, where the ruling grade could be approached on a good run, was put in freight service with 17-inch engines, and it did satisfactory work with the 17-inch engines, pulling over rugged parts of our division, until one day the train happened to stop at the foot of this ruling grade. That day the engine stalled, and it stalled simply because it was not as strong as a 17x24-inch engine when pulling at slow speeds. At very high speeds it was stronger."

GOODSPEED VALVE.

A cut-off valve for locomotives has been invented by Mr. William Goodspeed of Bloomfield, Iowa. This device consists of a plain slide valve having no outside lap, a loose cut-off plate encircling the valve and moved by the action of the valve.

The following description of the device is given in the words of the inventor. We quote as follows:

"This invention is in the nature of an attachment that may be applied to the slide valve of any ordinary engine with only slight modifications of the valve, and it is especially designed for use in connection with a link motion for controlling the speed and direction of rotation of the engine.

"My object in this invention is to provide a valve of this class, of simple, strong, durable and inexpensive construction, that may be adjusted, or set to automatically

cut-off at any desirable point, without employing any valve stem, or the like in addition to the ordinary ones used.

"A further object is to provide a valve in which a comparatively large and free induction-port is provided when the valve is set to cut-off at a relatively small portion of the piston's stroke to thereby tend to produce a high initial pressure and the consequent increase in the efficiency of the engine.

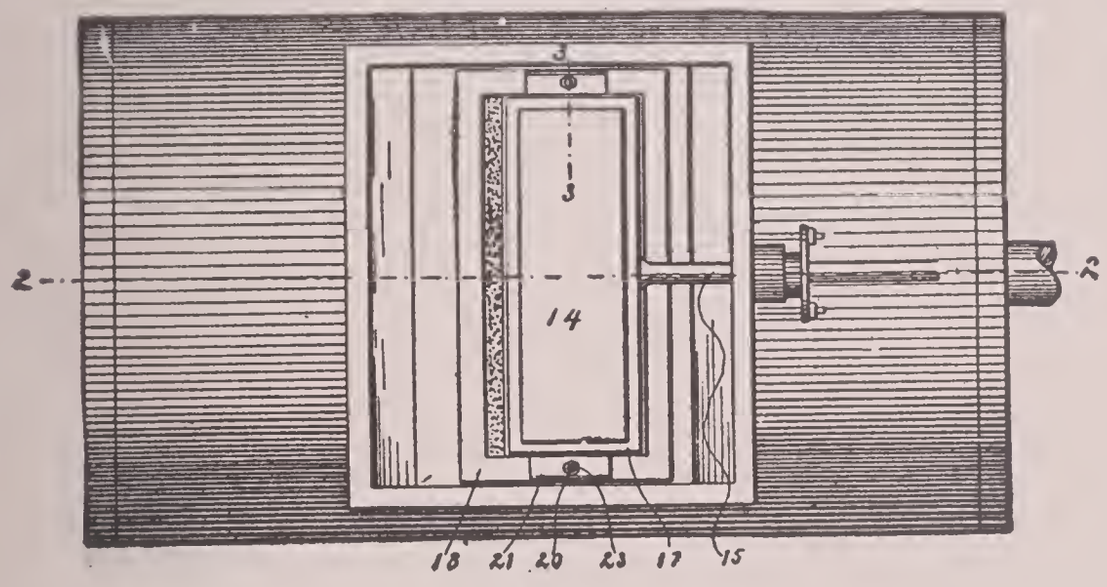


Fig. 51.

"A further object is to provide a valve which may be operated so as to admit steam during the entire stroke of the piston, so that the entire boiler pressure may be used throughout the entire stroke of the piston, when it is desired to generate an unusual amount of power, and further to provide an engine of this class in which the wear upon the valve seat will be extended over a large area, and hence the durability of the engine will be increased.

"Fig. 51 shows a top or plan view of the valve mounted on its seat within the steam chest.

"Fig. 52 is a central longitudinal vertical section on line 2, 2 of Fig. 51.

"Fig. 53 shows a transverse sectional view through the line 3, 3 of Fig. 51, and Figs. 54-55 show detailed perspective views of parts of the engine and valve to illustrate certain details of construction."

In describing the cuts, numbers, instead of letters will be used to designate the different parts.

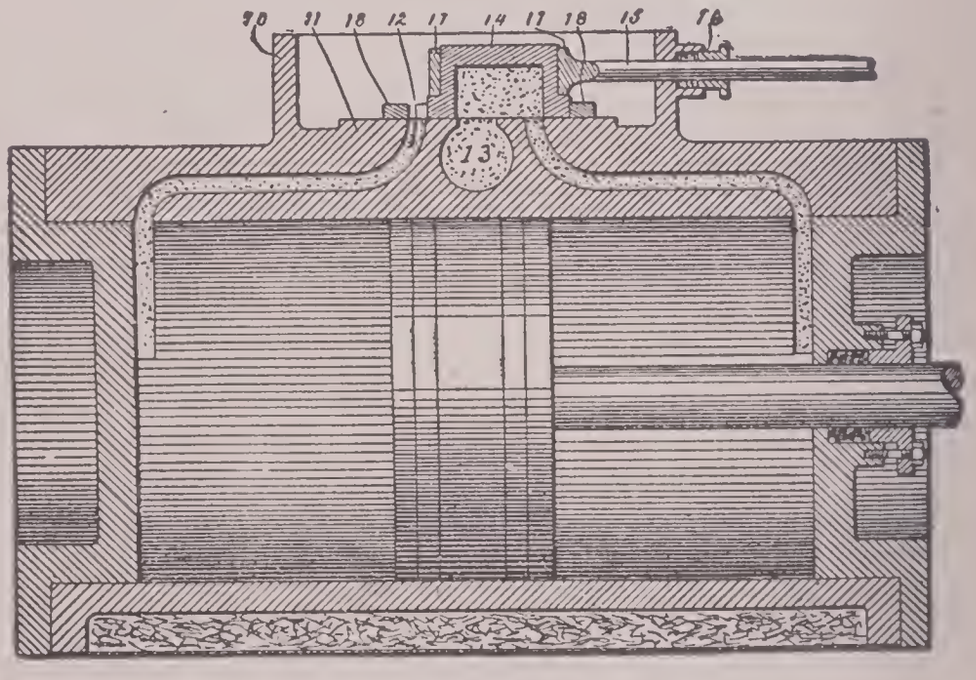


Fig. 52.

Referring to Fig. 52, 14 is a slide valve which differs from the valve of ordinary construction in that it has a diminished inside or outside lap, and 15 indicates the valve stem passing through the packing box, 16, and having a yoke, 17, (Fig 54), attached to it, which encircles the upper portion of the valve.

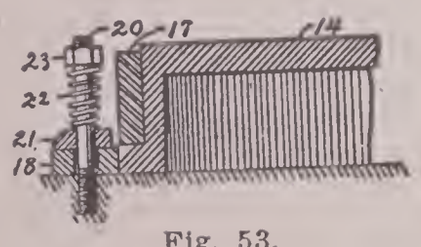


Fig. 53.

All of the above mentioned parts are of ordinary construction with the single exception noted.

The essential novelty in the valve consists of a rectangular auxiliary cut-off, 18, Fig. 55, having a flat lower surface, and having a rectangular opening in its central portion.

The dimensions of this central opening are in one direction, the exterior length of the valve plus the width of one of the ports, 12, and in the opposite direction exactly the width of the valve.

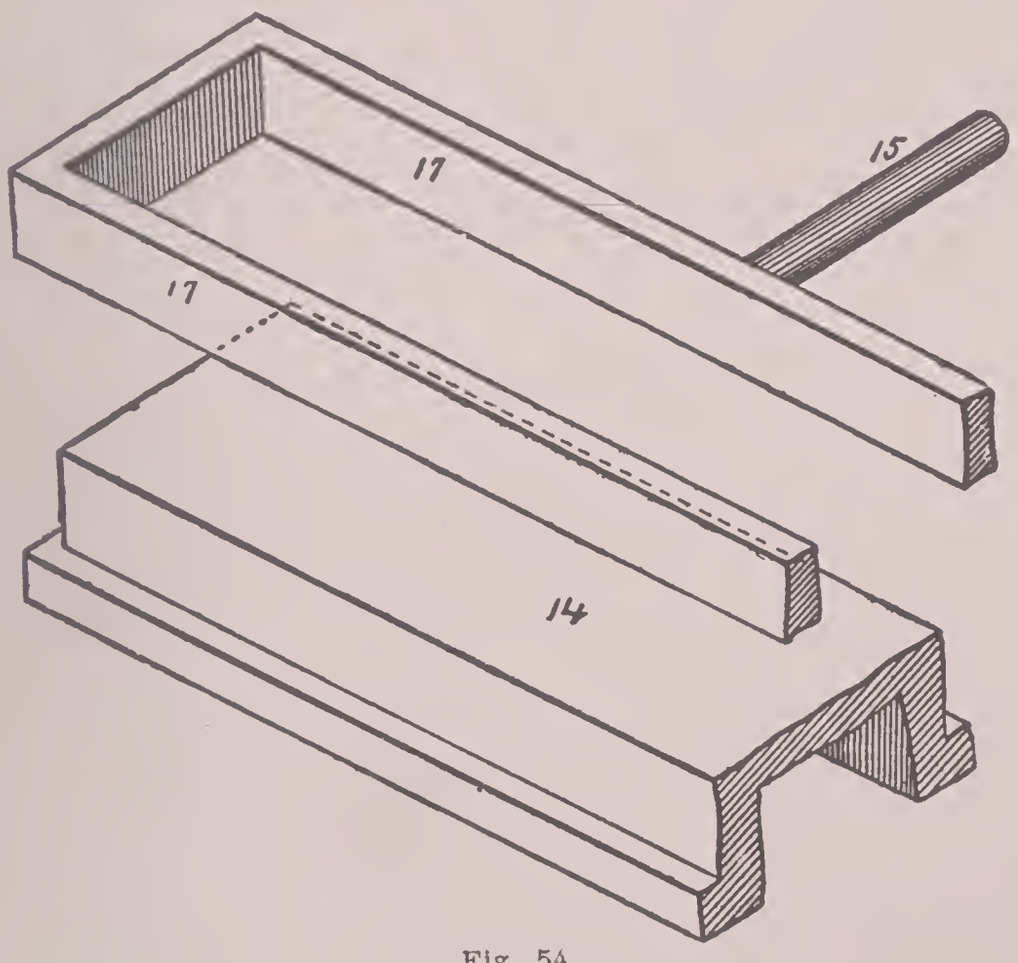


Fig. 54.

In the side pieces of the auxiliary cut-off are the longitudinal slots, 19, through which standards or guides project, being screwed into the valve seat.

21 indicates elongated washers having openings through which the standards, 20, are passed, so that the

under surfaces of the washers may engage the top surface of the auxiliary cut-off.

22 indicates spiral springs which are mounted upon the standards and compressed by nuts, 23, thus holding the washers, 21, in frictional contact with the auxiliary cut-off, 18.

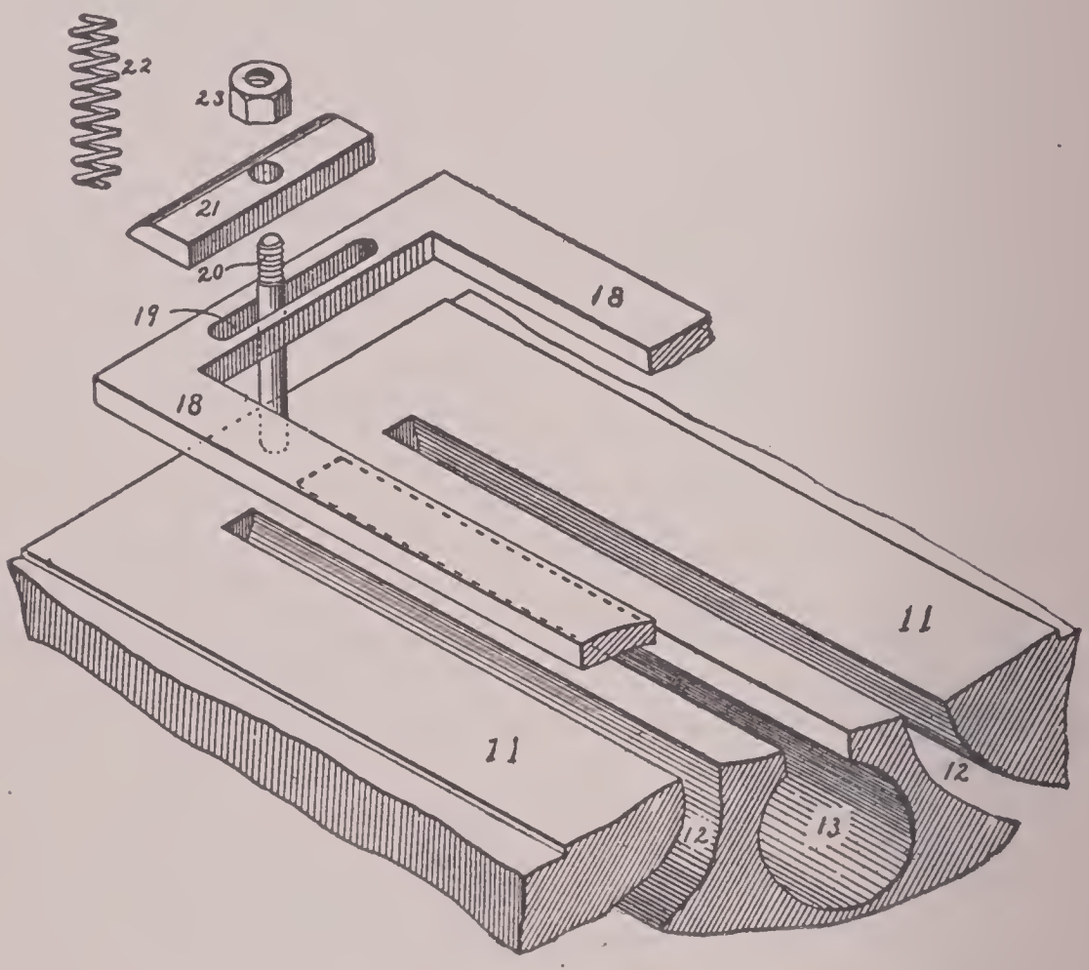


Fig. 55.

This arrangement is for use only when there is no steam pressure in the valve chest.

In practical operation it is obvious that the valve proper will be moved a slight distance upon its seat before engaging the auxiliary cut-off and when it does engage, the cut-off will be moved slightly, that is, when the

stroke of the valve is of greater length than the width of the ports leading to the cylinder.

This valve is designed for use only in connection with a link motion or other means of regulating the length of its travel.

Assuming that the link motion is set so that the valve will cut off at one-quarter stroke, and assuming the valve to be in its position for starting, that is, with a small lead opening, it is obvious that the first movement of the valve will be to open the induction port wide.

Then the auxiliary cut-off is moved by the valve so that the side of the cut-off adjacent to the induction port will be moved to partially cover the said induction port.

Then as the valve starts on its return movement it will meet the auxiliary cut-off, or that portion thereof that is partially covering the induction port at a point midway between the sides of the induction port, thus cutting off the steam while the valve has only partially covered the induction port, that is, before the full stroke of the valve has been made.

It is obvious that the further movement of the valve, that is, to the limit of its travel, and part of the way back, will not open the induction port.

Hence, as Mr. Goodspeed expresses it, "We have a one-quarter cut-off with a full opening of the induction port, and also a full opening of the exhaust port."

It is to be understood in this connection that the earlier the cut-off the greater is the required length of the valve travel.

When it is desired to work the engine to its fullest capacity the valve may be assumed to be in the same position as in the former instance, that is, with a small lead open.

Assume, further, that the link motion is set or adjusted so that the stroke imparted to the valve will be only the same as the width of the induction port.

It is obvious that in this instance the induction port will be opened gradually until at the end of the valve's stroke it will be wide open.

In this instance it will be noticed that the auxiliary cut-off is not moved, hence the friction of the valve upon its seat will be lessened, inasmuch as the area of the valve subject to the steam pressure is less than with the ordinary sized valve having considerable outside lap.

HALEY'S SLIDE VALVE.

The objects sought to be attained by the use of this valve are an increased expansion of the steam after cut-off, and the relief of excessive compression.

It is the invention of Mr. J. A. Haley, of Fort Wayne, Ind.

It consists in the construction and combination of, first, a main valve constructed with shortened face, but adapted to actuate an independent valve face, the object of which is to supplement the shortened valve face.

Second, an independent valve face, and, third, devices and means whereby during a portion of the travel of the main valve, the independent valve face is not operated by the main valve, but remains stationary, and thus serves to give an increased inside lap at one part of the valve, together with increased clearance at the other part alternately with each stroke of the main valve.

By the term "independent valve face," is meant a valve face which during a given portion of the travel of the valve remains stationary and is not moved by the valve either directly or indirectly.

The form of construction of the Haley valve is shown in Figs. 56 and 57, where A is the main valve, which has portions of its faces cut out in the form of a rabbet, thus leaving the faces, a, shortened.

These rabbets are cut out sufficiently deep to permit an independent valve face, B, to be placed underneath them and afford room for its independent motion.

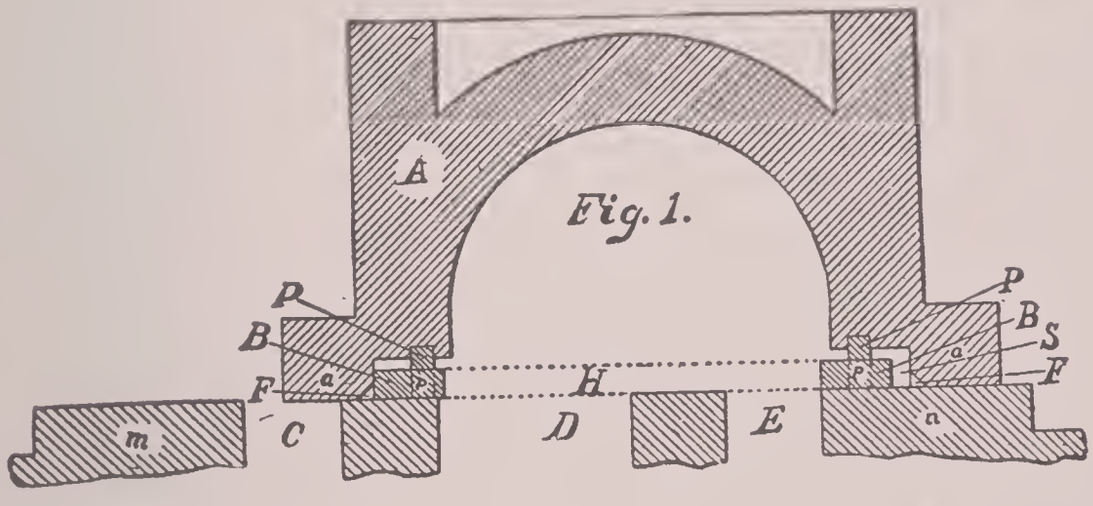


Fig. 1.

Fig. 56.

This independent valve face consists of two bars, B, B, connected by cross bars, H, which are also placed under rabbets on either side of the main valve, A, and its dimensions are such that there will be a space, S, between it and the extreme length of the rabbets on the inside of the face of the valve, A.

This frame composing the independent valve face is placed on the valve seat within the rabbeted parts of the face of the valve, A, and packing, P, is placed above the independent valve face, and other packing, p, is placed upon the side, in the side rabbets, as shown in Fig. 57, for the purpose of preventing steam from passing between the valve faces, B and a.

The space S is for the purpose of, and designed to allow a movement of the main valve, A, through that distance or space, without moving the independent valve face, B.

Its length is determined by the requirements for inside lap, and inside clearance.

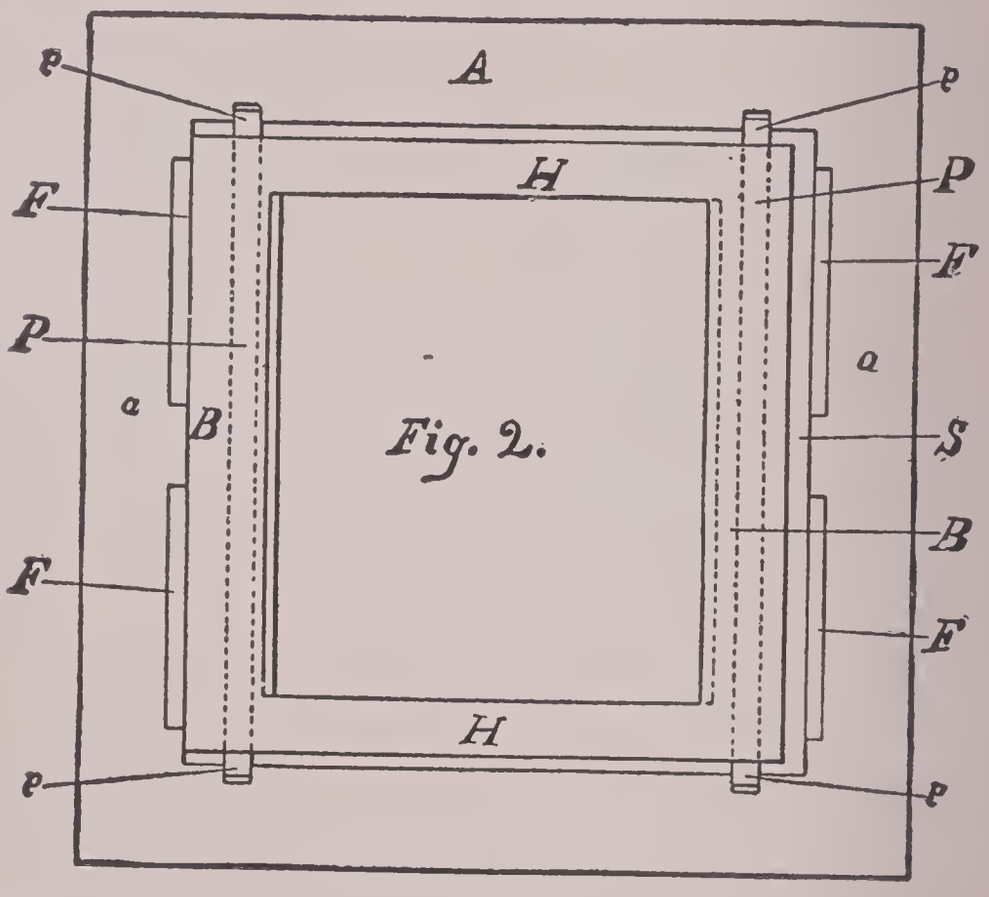


Fig. 57.

When the space S has been traversed by the valve A moving from n to m, the end of the rabbet impinges against the independent valve face B at the port E and moves it to the end of travel of main valve A.

On the return stroke the pistons and movements are reversed. The faces, A, have chamfers, F, on their inside corners for the purpose of permitting the steam to

operate against the ends of the independent valve face, B, to aid in keeping it alternately pressed against the ends of the rabbets for the purpose of preventing any movement of B, except when actuated by the main valve, A.

The operation of the device is as follows: When the engine is first started the ports are used to their full capacity, steam being admitted to the cylinder throughout the stroke of the piston, in which case this valve has no special operation, but when the stroke of the valve is reduced or shortened in order to use steam expansively then the following results take place:

Referring to Fig. 56, let it be assumed that the opening of the port, C, as shown, is the extreme opening for a given point of cut-off.

At this point the cylinder is therefore taking the full amount of steam at such point of cut-off, while the port, E, is exhausting at the other end and is open to its full extent.

The width of the exhaust cavity measured on the valve seat is determined by the fixed distance apart of the inner edges of the two plates, B, B, which form the independent valve face.

The packings, P, p, prevent the passing of steam between the faces, B and a, into and from the exhaust cavity.

The area of space on the valve seat covered by the combined valve faces, B and a, varies during the travel of the valve.

When the valve travels, say, on the return stroke (see Fig. 56), the faces, B, B, remain stationary until the distance, S, has been traveled by the face, a, at the end, n, of the valve seat.

The inner end of the face, a, then impinges against and actuates B.

At this point of travel, and continuing to the end of the return stroke, the combined faces, B and a, on the end, n, of the valve seat, cover the diminished distance, while the faces, B and a, on the other end, m, cover an increased distance over the ports and valve seat, the difference being represented by the space S.

Such movement increases the inside lap of the combined valve faces, B and a, on the end, m, of the valve seat, because of the lengthening of the combined valve faces, and thereby delays the release of the steam at the port, C, thus allowing a greater expansion before exhaust takes place.

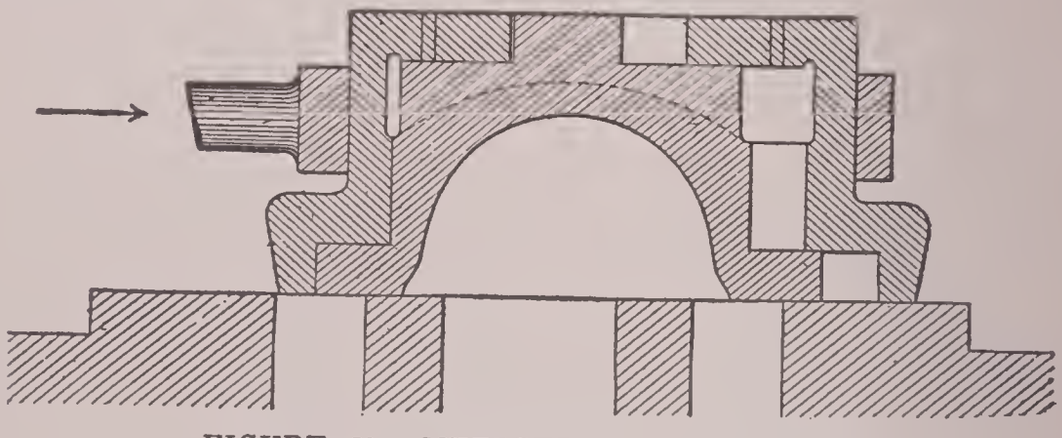


FIGURE 58. METZGER'S SLIDE VALVE.

At the same time while so traveling, the distance covered by the combined faces, B and a, at the end, n, has been reduced correspondingly, and the face, B, has remained stationary during a portion of the travel, so that the closing of port, E, has been delayed, thereby giving a free release and reducing the compression in that end of the cylinder.

This increase of the inside lap at the end, m, of the valve seat and increased clearance at the opposite end, n, is reversed with the reverse travel of the valve.

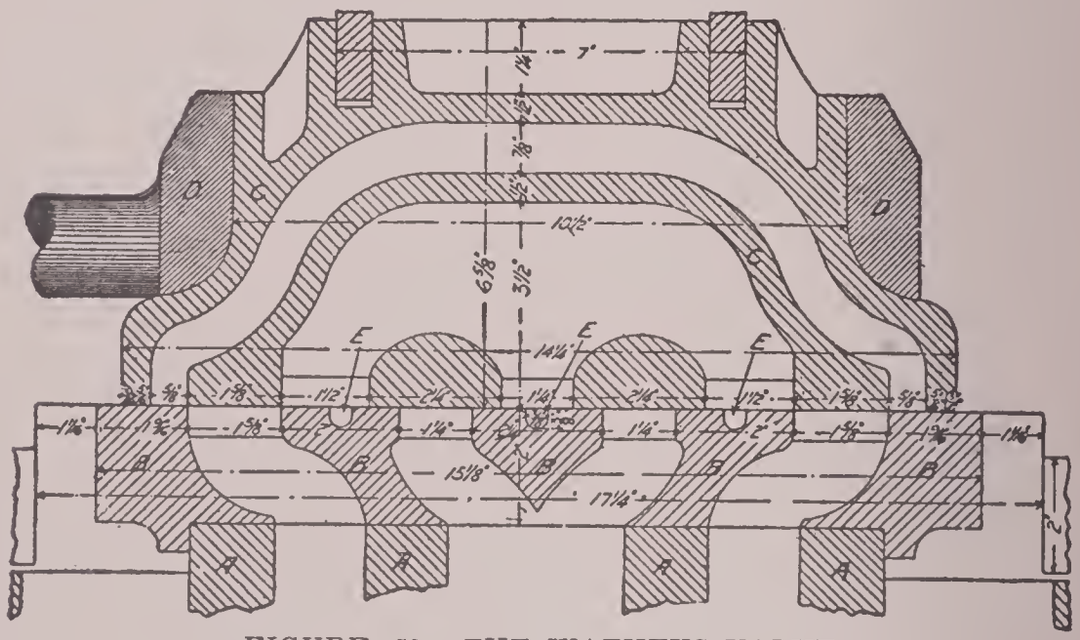
METZGER'S SLIDE VALVE.

The principle involved in the design of this valve is practically the same as that in the Haley valve which has just been explained. Fig. 58 is a vertical section through the valve and shows the construction plainly, the object being the same as with the Haley valve, viz., an increased inside lap and free release.

This form of valve is the design of Mr. Jules P. Metzger, of Paterson, N. J.

A NEW LOCOMOTIVE VALVE AND VALVE SEAT.—C. I. & L. RY.

There is shown in Fig. 59 a valve and valve seat which, while tending in a direction the limit of which has probably been tested in another design, is, we believe, quite new. It should be understood at first that the drawing shows the valve as applied to the ordinary cyl-


7"
1/4"
2 1/2"
7/8"
1 1/2"
10 1/2"
6 5/8"
3 1/2"
14 1/4"
E
E
E
5/8"
1 1/8"
1 1/2"
2 1/4"
1 1/8"
2 1/4"
1 1/2"
1 1/8"
5/8"
1 1/4"
1 1/4"
1 1/4

valve seat and steam chest would be of the same height as in the ordinary design.

The valve is shown as central on the seat, and the exhaust edges of the steam ports are line and line with the inside edges of the valve, so that if it is considered that the valve is moving from right to left, the left steam port is about to open for exhaust. The double exhaust port, however, is covered ½ in., so that after the point where the interior of the valve is opened to the exhaust steam, the valve must travel ½ in. before the exhaust port begins to open. The exhaust port being double, the area of the port which is uncovered by a certain movement of the valve is twice the area that would be uncovered were the port single, and a freer exhaust must therefore result. The double exhaust port and single steam port would be no better than if both ports were single were it not for the arrangement which makes it possible to open the exhaust side of the steam port ½ in. by the time the valve begins to uncover the exhaust port. The claim made for the valve is that the steam can be held in the cylinder during a longer portion of the stroke (except for the steam which expands into the inside of the valve), because when the valve begins to uncover the exhaust port the required opening is given by one-half the travel of the valve required for the single exhaust port. The grooves E E E are cut in the seat to assist lubrication; it is reasoned that they are soon filled with water or oil, or a mixture of both, and that the face of the valve is coated each time the valve passes over them. The valve is the design of Mr. H. Watkeys, master mechanic of the C. I. & L. Ry.

One of the greatest drawbacks to the efficiency of high speed locomotives is excessive compression.

This is a fact that cannot be questioned, and the device which will distribute the steam to the pistons, and from thence to the atmosphere with the least loss of heat, while at the same time smooth running is effected, will always meet with favor.

Not only does great loss of heat occur through compression, but another source of loss is excessive clearance, whereby a needlessly large volume of steam is required to fill the space between piston and cylinder head at each end of stroke.

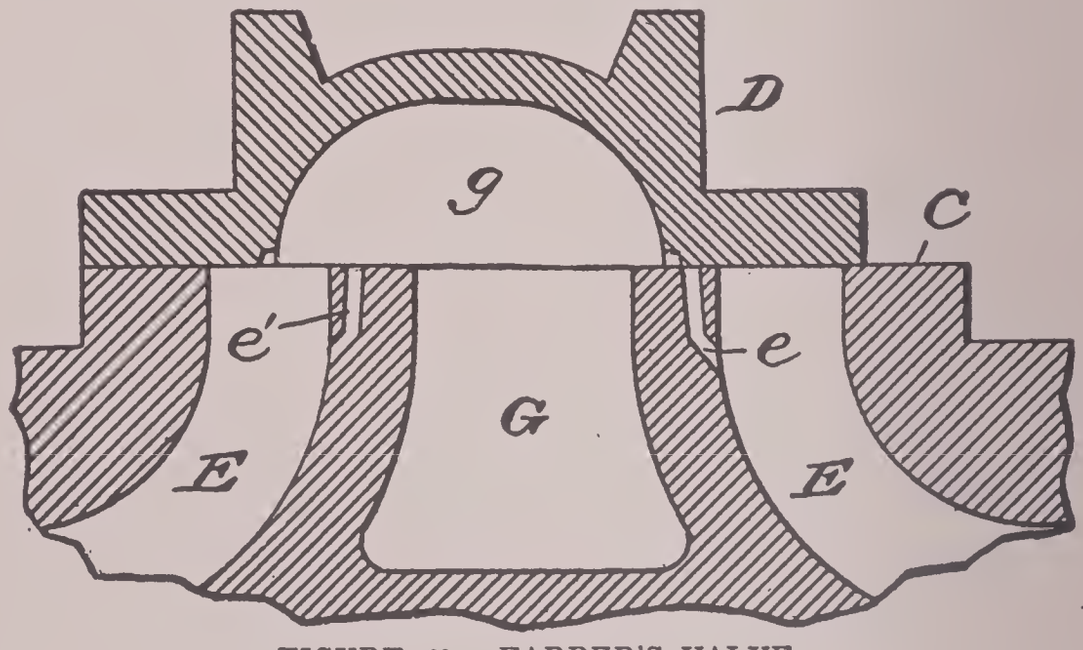


FIGURE 60. FARRER'S VALVE.

FARRER'S VALVE.

Fig. 60 illustrates the improvement of Mr. Chas. S. Farrer of Dunmore, Pa., as applied to the ordinary slide valve and the principle of the device is as follows:

Two small holes, e, é, are drilled through each bridge in the manner shown.

These serve as relief ports, and coincide with the grooves cut on the exhaust edges of the valve D, thereby

permitting a portion of the air that is being compressed in the cylinder to escape to the exhaust cavity G and thence to the stack, thereby reducing compression.

The advantages obtained by this arrangement are as follows: First, a smaller clearance space can be made. Second, higher speed can be obtained owing to the fact that excessive resistance in the front of the piston, in other words back pressure, is obviated, when cutting off at short points of cut-off, and third, a saving in the general wear of the engine, and especially in the wear of the valve and seat, by reason of the valve keeping down on its seat at all times and not being forced off its seat by back pressure caused by excessive compression.

There is therefore a saving effected in the coal consumption of the engine, as steam cannot blow under the valve to the exhaust port and the engine will do the same work at shorter points of cut-off.

BALANCED SLIDE VALVES.

For a great many years the plain slide valve answered all the requirements for locomotive service, but with the enlargement of the locomotive and the increased steam pressures it was found almost impossible for one man to reverse an engine with such an enormous pressure on top of the large slide valve which modern locomotives require. The object of designers was, therefore, to produce a valve that would require as little power as possible to move it. To lessen this friction, what is known as the "roller valve" was first invented. It was a plain slide valve with rollers attached to each side of the valve. While this valve required less power to handle it, yet it failed to remove the cause of the great amount of friction between the valve and its seat, and therefore did not come into general use, although it is still in use upon some roads. But when the balanced valve, sometimes called the equilibrium slide valve, was invented, the correct principle, seems to have been followed, viz.: the removal of the steam pressure from the back of the valve. These valves have been universally adopted. The various forms of these valves that have been invented are too numerous to mention, but they are all constructed upon the same principle, that of removing the pressure off the valve.

We have illustrated a few forms of these valves, varying all the way from very good to very bad. The best form of balance valve will be found to be the one which overcomes the greatest friction (within practical

limits), the most simple in construction and positive in its operation and has the fewest parts which are liable to break or get out of working order.

THE GOULD BALANCED SLIDE VALVES.

Figures 61 and 62 serve to illustrate the construction and operation of the Gould balanced slide valves, of which there are two styles.

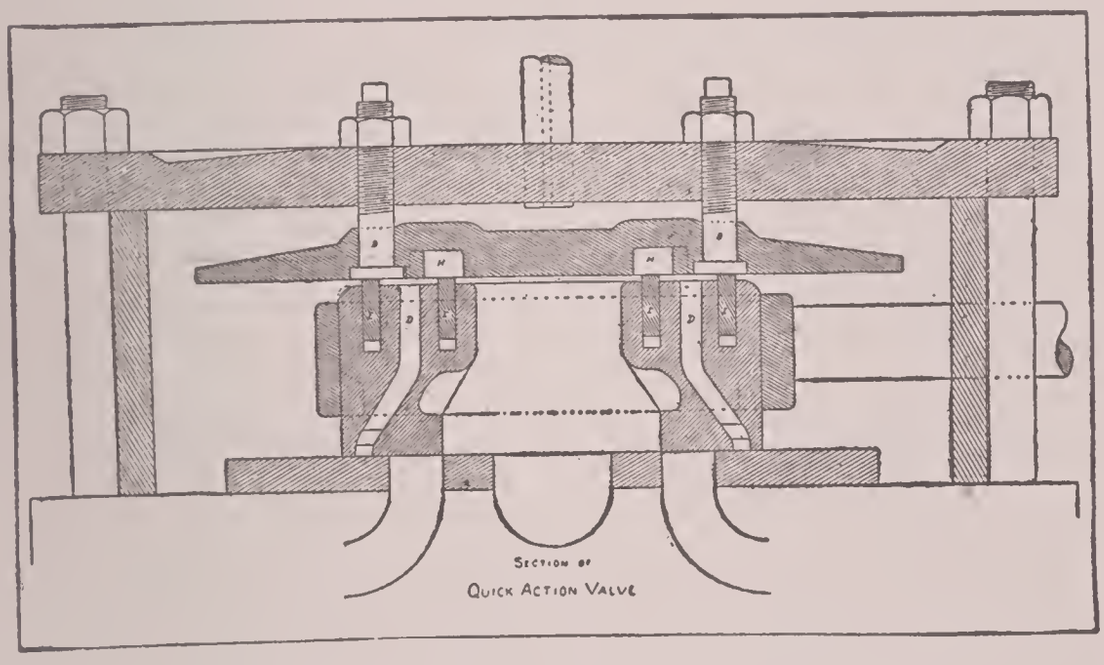


FIGURE 61. GOULD QUICK ACTION VALVE.

These valves are the invention of Mr. W. F. Gould, of Des Moines, Ia.

The form known as the "quick action valve," Fig. 61, deserves special attention.

Heretofore it has been the generally accepted belief among practical men that it would be impossible to prevent the valve from lifting off its seat if all the pressure were removed from the top of the valve, but the action of the "quick action valve" appears to refute this theory.

The principles of the Gould balanced valve are similar to those of the piston valve.

Either form of the Gould valve may be applied to the ordinary flat valve seat.

Two methods of balancing are employed, one form of which, the "quick action valve" may be used with either a flat pressure plate, or a semi-circle balance plate, and promises to be an important rival to our best forms of balance valves.

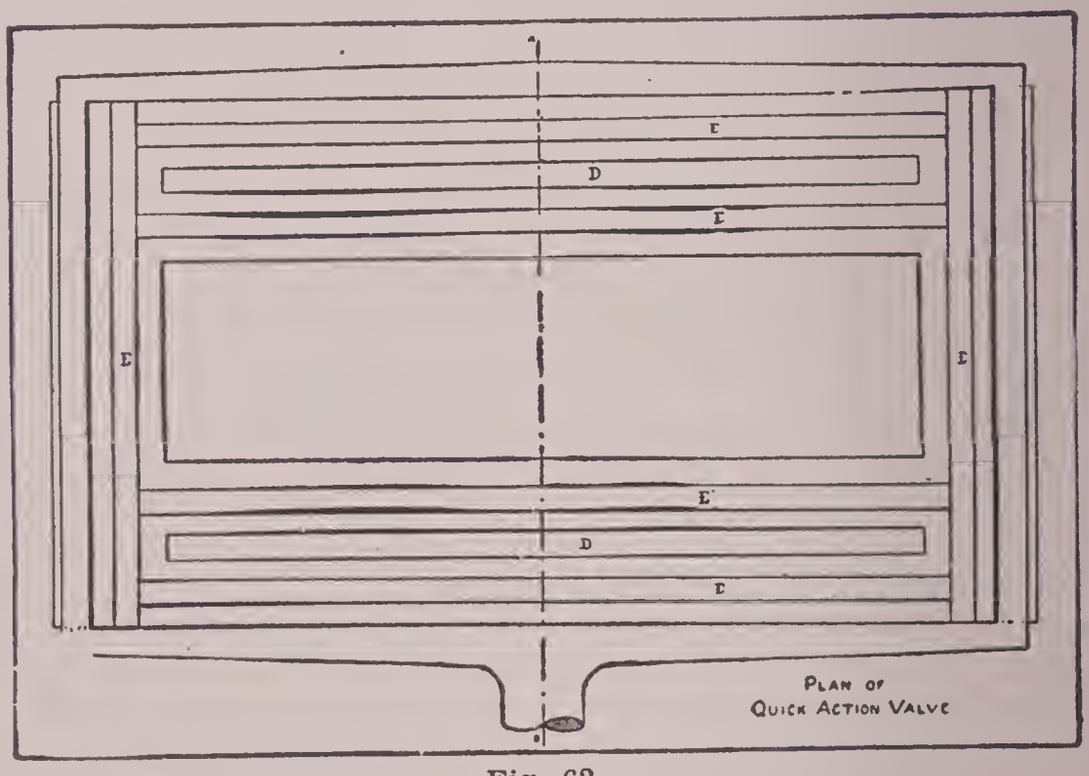


Fig. 62.

For the other form of valve the balance is obtained by means of a semi-circular balance plate fitted into the steam chest lengthwise.

It is not bolted to the cover like most pressure plates, but rests on the valve seat. It has lugs that are closely fitted on the ends to prevent it from moving lengthwise, while the pressure on its back serves to hold it to its seat.

The top of the valve is also a semi-circle in form, slightly smaller than the plate, and has balance strips set into it which bear against the plate in the usual manner.

A small port at each end permits live steam to enter between the valve and the pressure plate, which is permitted to cover sufficient area to overcome the back pressure from the cylinder, thereby obtaining almost a perfect balance.

In construction the two balance plates, (one for each valve) are first planed off on the edges, then the two are clamped together and bored out to their required size.

The valves are finished in a similar way, first the two faces are planed off, then clamped together and turned off in a lathe, and made a little smaller than the pressure plate.

When the valve requires facing, the same amount is taken off the bottom edges of the pressure plate also, thereby retaining their original positions, and the plates automatically adjust themselves to any equality due to the wear of the valve.

Fig. 61 is a sectional view of the quick action valve, and shows the relative position of the small ports D, D at either end of the valve, to the ports leading to the cylinder. (Not lettered.)

The valve cover A is supported by four bolts to the steam chest cover, in such a manner as will permit of the cover A to be raised or lowered at will. H, H are ports in the base of the cover running longitudinally parallel to the steam ports.

The ports D, D already referred to are longitudinal cavities in each end of the valve, and should be of the

same length as ports H, H and nearly as long as the steam port that leads to the cylinder.

E, E are packing bars placed on each side of the cavity D in the top of the valve in such a manner that the distance between them is of very little more area than the width of the steam port, for the purpose of holding the valve down when the steam port at the end of the valve is filled with steam.

In the operation of the valve as shown herein, when the valve is pulled back so as to open the steam port, it will be observed that the packing bar nearest that side will have also been pulled back so that the steam at the top of the valve will pass over that top of the packing bar E into cavity H, thence through cavity D, into the steam port, thereby forming a double opening into the cylinder.

Another feature shown in Fig. 61 is that the valve on its return shows the port D in communication with the steam port after cut-off has occurred, thereby showing the area between the packing bars E, E to be in communication with the steam in the steam port. This is done in order to balance the valve.

It should be noted that the outside packing bar can be placed as near the outside end of the valve as desired, so as to take all, or nearly all, of the pressure off of the valve.

Another noteworthy feature connected with this valve is, the relative position of the ports to each other when the engine is running at a high rate of speed, and a comparatively short cut-off, or in other words, using the full amount of lead.

In such cases there is more or less cushion, or back pressure in the cylinder, but owing to the fact that port

D will be in communication with the steam port, the bad effects of this back pressure are entirely obviated.

When it becomes necessary to reverse the engine while running at a high rate of speed it will be observed (by studying Fig. 61), that the valve will lift, and relieve the pressure in the cylinder, for the reason that the movement of the valve will be exactly opposite to the movement it had before the engine was reversed.

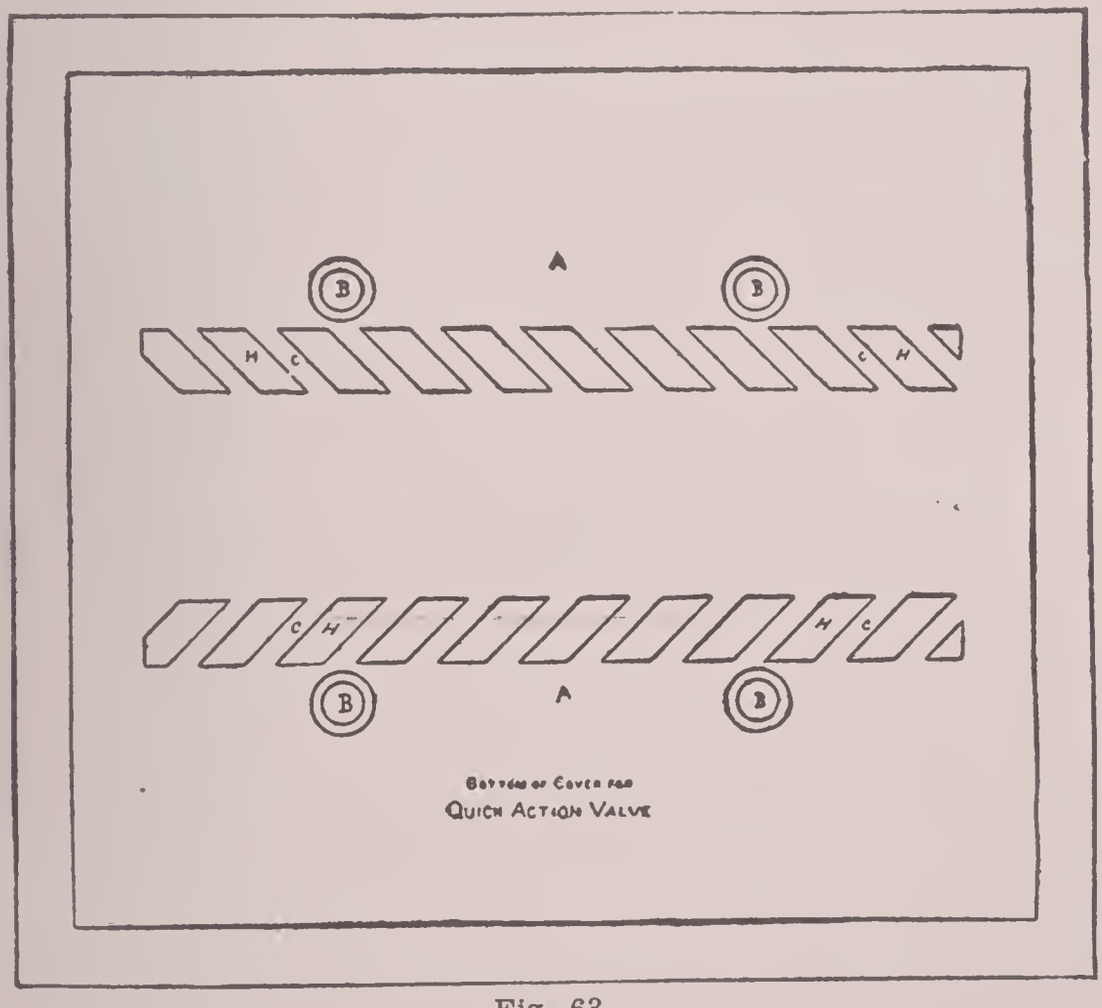


Fig. 63.

In the plan view (Fig. 62), the relative position of the side bars and the end bars E to each other is clearly shown.

The valve is also shown to have an open back, which prevents the exhaust steam from having any effect upon

the valve, but allows it to keep the bottom of the cover well lubricated.

Fig. 63 shows a view of the bottom of the cover A. The ports H, are shown placed diagonally with the small bars C, in them.

The reason that ports H are placed diagonally is to prevent ridges from becoming worn on the packing bars E as they move across ports H.

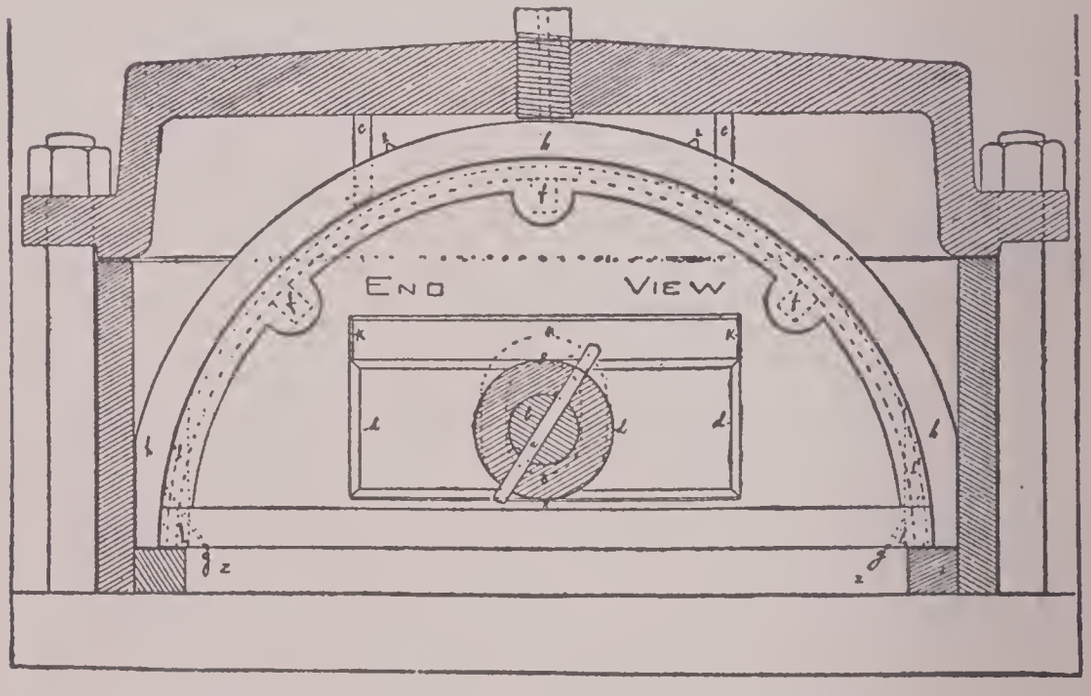


FIGURE 64. GOULD BALANCE VALVE.

Figs. 64 and 65 are sectional views of the Gould balance slide valve, Fig. 64 being an end sectional view of the valve, valve cover and steam chest, and Fig. 65 is a vertical side sectional view of the same.

Referring to Fig. 64 H, is a loose cover that fits over the top of the valve, and is prevented from having too much longitudinal movement by stops c, c. This movement should never exceed 1-16 of an inch over all.

The space occupied by the packing is designated by the dotted lines in the semi-circle, and this packing is enclosed in the pockets f, f, f.

The letters k, k, represent a chipping strip upon the rear end of the valve, which may be faced off so that the back end of the yoke d, may fit it.

The letters n, n, represent the sleeve, which is cast solid to each wing of the valve, and extends across the cavity o, in order to contain the valve stem b.

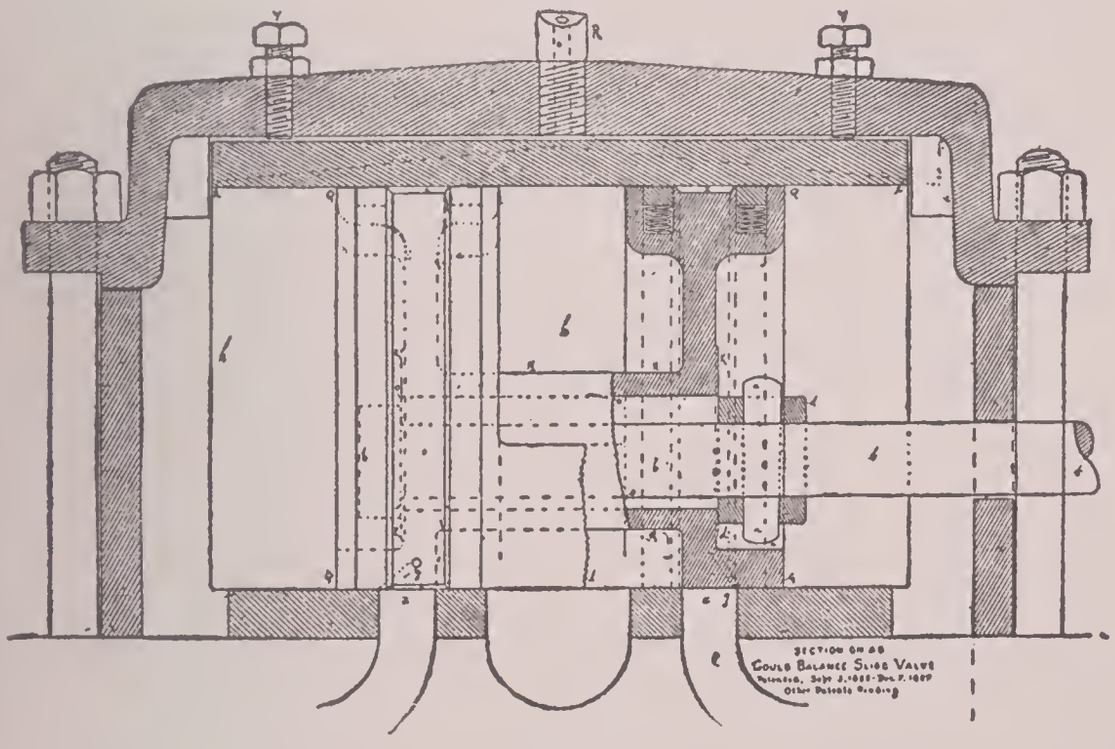


Fig. 65.

The cavity o, is oblong in shape in order that the vertical movement of the valve may not bind upon the valve stem.

The letters g, g, Fig. 64 designate two small holes drilled in the face of the valve for the purpose of establishing communication between the port z, and the cavity x, shown in Fig. 65.

Small ribs e, e, are cast upon the top of the valve cover h, in order to prevent the valve oil from running down the sides of the cover.

These ribs e, e, also cause the oil to be distributed to the end of the cover, thereby effecting a more perfect lubrication of the valve and seat.

Fig. 65 is a side sectional view of the valve, in which the packing rings are shown, held up by the spiral springs on one end of the valve, while the dotted lines on the opposite end of the valve designate the spring pockets.

Two cavities x, x, Fig. 65 extend around the outside faces of each end of the valve, while the packing as shown is placed upon each side of these cavities x, x.

The area of cavity x should always slightly exceed the area of the steam port z, in order that when port z, is filled with steam, and communication established with cavity x through hole g, (shown also in Fig. 64) the pressure in cavity x will at least equal the port pressure, and the valve will be held down on its seat.

It will be observed that a practically perfect balance may thus be obtained, the degree of perfection depending upon the location of the packing rings, relative to the outside edge of the valve, and the giving of the proper area to cavity x.

The check stubs y, y, Fig. 65, in the top of the steam chest cover are for the purpose of regulating the vertical movement of the valve cover h and preventing it from lifting too high from the valve.

It will be observed that these stubs do not quite touch the valve cover, being adjusted to allow a slight movement in a vertical direction.

Letters l, l, see Figs. 64 and 66, designate a web that holds the bottom of the wings of the valve firmly together

in one piece and prevents any leakage of steam into the exhaust, which might occur if the valve cover was raised from the seat.

This valve is semi-circular in form and is constructed in the following manner.

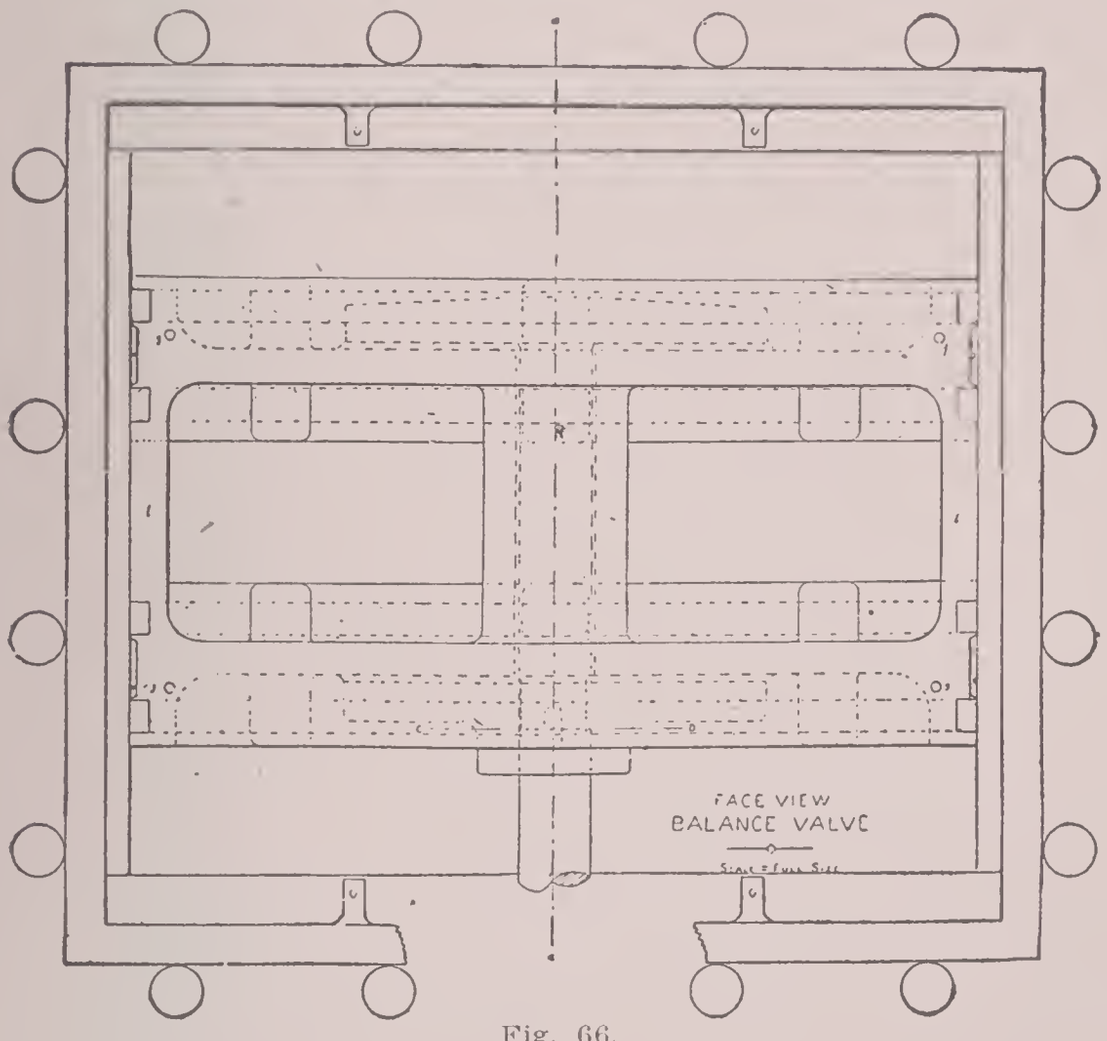


Fig. 66.

As soon as the castings come from the sand their faces are planed and fastened together, and they are then placed in a lathe and turned to the proper size.

Grooves for the packing, also cavities x are cut into their faces. The valve covers have their edges planed, and they are then fastened together and bored out to fit

the valves, thereby making one for each side of the engine.

The valves and their covers should fit iron to iron when they are cold, for the reason that when the cover becomes hot it expands a little more than the valve does, which leaves the valve free to move perfectly easy under it.

When it becomes necessary to face the valve the same amount of iron should also be planed off of the bottom edge of the valve cover, thus causing each to assume their normal position relative to each other, and as cavity o in the valve through which the valve stem passes is made oblong vertically, it will readily be seen that facing the valve will not affect the valve stem in the least.

When the exhaust is being planed out the tool should not be allowed to cut through line l shown in Fig. 66, which is a face view of the valve, and shows the form of the exhaust.

The sizes of the ports will of course have to be governed by the sized engine upon which the valve is to be used.

The packing rings should have at least five-eighths of an inch face with a depth of one-half inch.

The valve stem should be at least 1½ inches in diameter, and the stops c, c, Fig. 64, should not be less than ⅝ of an inch in width.

The valve cover should finish 1¼ inches in thickness, in order that there will be no danger of its ever becoming distorted, or out of shape. This thickness would also permit of it being bored out the second time to fit the valve.

The depth of the cavities x should not be less than ⅛ of an inch and the diameter of the hole g, should be not less than 3-16 of an inch.

The diameter of the bolts y should be not less than ¾ of an inch.

The springs that are designed to hold the packing should be made of about No. 70 steel wire so that the pressure on them would be very light for the reason that if these springs are too stiff they will cause the packing to wear too fast or might hold the cover up from the valve when steam was not being used.

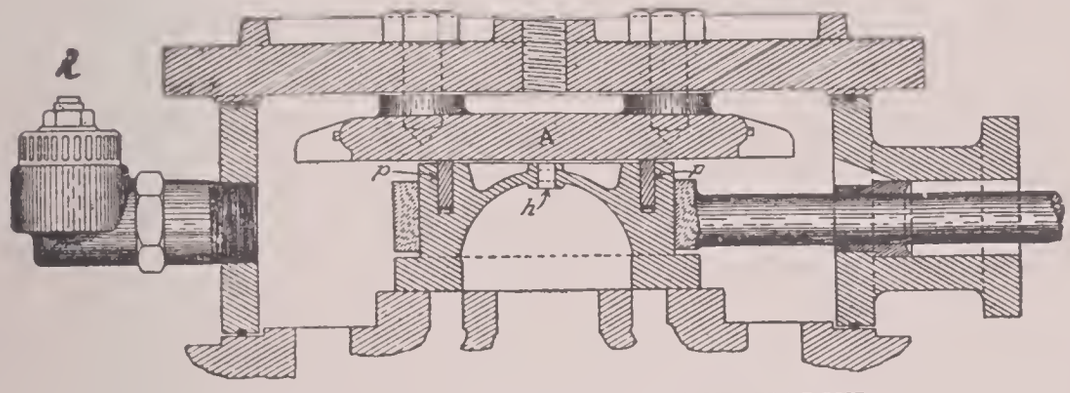


FIGURE 67. VACUUM RELIEF VALVE.

VACUUM RELIEF VALVE.

This valve, R, Fig. 67, is designed to be placed in the steam chest to automatically supply clean air to the cylinders through the air valve when engine is running shut off, and thus furnishes a free supply of air from the outside instead of its being drawn in from the smoke box laden with hot gases and cinders which lap all oil from the valves and seats.

The pressure relief valve performs a very valuable function in preventing the dangerous accumulation of pressure in the steam chest and dry pipe and often breaking of same when engine is suddenly reversed. The valve is set to open at a pressure slightly above the maximum boiler pressure, and will allow any excess of pressure to escape to the atmosphere, yet will maintain in

the cylinder a uniform pressure of air within the limits of safety, when running forward after reversing, and thus supply resistance to the piston and overcome the momentum of the train, and perform the functions of an automatic air brake in assisting to stop the train. By using this valve an engine may be suddenly reversed while running at high speed without strain or damage to any portion of the machinery or boiler.

This valve should be well designed and made from the best steam metal. Combined pressure and vacuum relief valves on low-pressure steam chest and single-pressure relief valves on low-pressure cylinder heads should be set at 45 per cent of the boiler pressure, and the high-pressure cylinder head relief valves set at 20 lbs. above boiler pressure.

THE PISTON VALVE.

Reference has already been made to the piston valve for locomotives and as there are many different varieties, of this type of valve, the discussion will be continued for a short space.

The piston valve is an old affair in locomotive practice, having lain dormant for years. One of the early designs of these valves was that of Mr. Thomas S. Davis of Jersey City, in 1866. A difficulty which developed with his design and others at that time was the rapid wearing away of the valve cage at the port openings, due in part to the absence of bridge strips in the port openings, as ring-bearers for the piston. It was supposed also that the valve piston rings needed adjustment the same as those of the steam cylinder piston. Occasionally this adjustment was faulty and, cramping the free motion of the valve, over-balanced all the advantage of the piston for

the time being. The tallow then used for lubrication troubled the piston valve, as it did also the plain D valve. The cause which more than any other, however, led to the disuse of the piston valve after these early experiences was the introduction of the balance on the slide valve. This balance rendered the slide valve form acceptable at the then low pressure standard on engines.

In the meantime the piston valve has gained a position at the head in steam vessels, small and large, naval and commercial, at all pressures the world over. It is used in fast steamers crossing the ocean, the entire distance being passed, in some instances, without lubrication. It is also used in the best and fastest electric engines running almost continuously. The piston valve has of late years been improved in form, eliminating the features which caused objections to the old form, and with the advent of high pressures it is fast coming into favor again.

Owing to the fact that the piston valve is practically encased within the walls of the cylinder it permits of a greater port area, and occupies less space than the D slide valve does.

It has been commonly supposed to be a perfectly balanced valve, but recent tests have proved that it is not as perfectly balanced as was generally supposed.

It has been found that its perfection of balance depends largely upon the width of its rings, and the steam pressure, for the simple reason that so long as the steam exerts a pressure under the rings, thus holding them against the walls of the cylinder, there is thereby created an unbalanced friction the amount of which depends upon the area of the rings, and the pressure of the steam under them.

Of course a great deal depends upon the workmanship

in the construction of the valve. The closer the rings are made to fit the grooves without binding, the better, but it is practically impossible to fit the packing rings perfectly steam tight without their sticking in the grooves, and the more the sides of the rings wear from service, the greater will be the steam pressure under the rings, and as a consequence the greater the friction.

Owing to the usual forms of piston valves, they being generally made either of one long casting, or in two parts connected with a rod or tube, it is possible to locate the two faces near the ends of the cylinder, thus reducing the clearance considerable, as compared with the D valve.

This lessening of the space to be filled with live steam between the piston and the valve face, for each stroke of piston, most certainly tends toward an increase in the efficiency of the engine, as the expansive force of the steam is in no way decreased, while the weight of steam to be exhausted at the completion of the stroke is considerably less. While the correct principle for reducing the clearance seems to be followed in placing these valve faces near the ends of the cylinder, it should be remembered that while the distance between the valve and piston is lessened, the clearance space extends clear around the valve, and the claim of reduced clearance for these valves has frequently been overthrown by the Indicator.

In the Vauclain Compound locomotive built by the Baldwin Co. the valve employed to distribute the steam to the cylinders is of the piston type, working in a cylindrical steam-chest located in the saddle of the cylinder casting between the cylinders and the smoke-box, and as close to the cylinders as convenience will permit.

As the steam-chest must have the necessary steam passages cast in it, and dressed accurately to the required

sizes, the main passages in the cylinder casting leading thereto are cast wider than the finished ports.

Fig. 68 shows the arrangement of the high and low pressure cylinders in relation to the valve. The steam-chest is bored out enough larger than the diameter of the valve to permit the use of a hard cast iron bushing

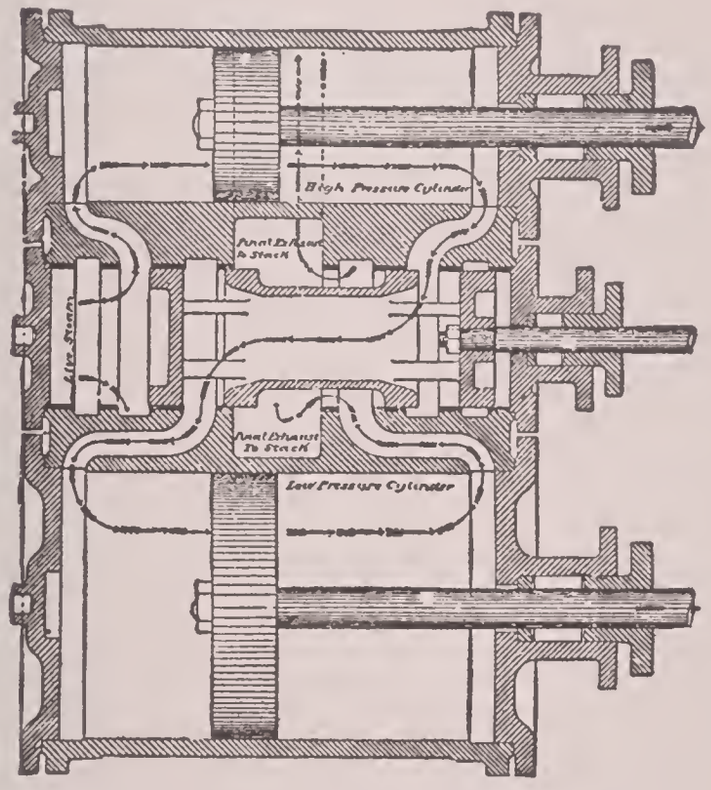


FIGURE 68. SECTION OF CYLINDERS AND VALVE CHEST—VAUCLAIN COMPOUND ENGINE.

(Fig. 69). This bushing is forced into the steam-chest under such pressure as to prevent the escape of steam from one steam passage to another except by the action of the valve. Thus an opportunity is given to machine accurately all the various ports, so that the admission of steam is uniform under all conditions of service.

The valve, which is of the piston type, double and hollow, as shown in Fig. 70, controls the steam admission and exhaust of both cylinders. The exhaust steam from

the high-pressure cylinder becomes the supply steam for the low-pressure cylinder. As the supply steam for the high-pressure cylinder enters the steam-chest at both ends, the valve is in perfect balance, except the slight variation caused by the area of the valve-stem at the back end. This variation is an advantage in case the valve or its connection to the valve-rod should be broken, as it holds them together. Cases are reported where compound locomotives of this system have hauled passenger trains long distances with broken valve-stems and broken valves, the parts being kept in their proper relation while running by the compression due to the variation mentioned. To avoid the possibility of breaking, it is the present practice to pass the valve-stem through the valve and secure it by a nut on the front end.

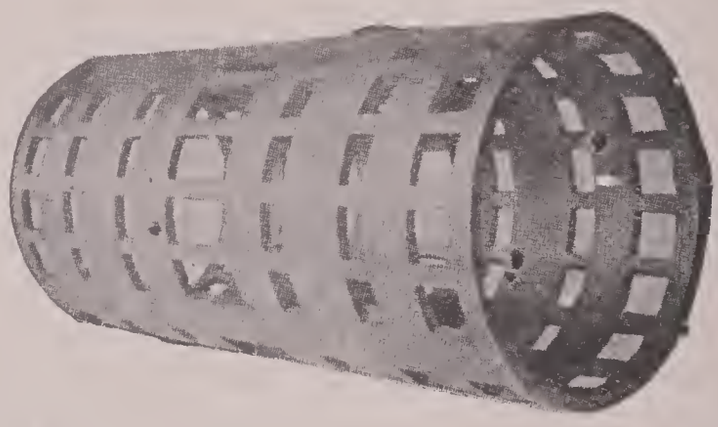


FIGURE 69. BUSHING FOR VAUCLAIN PISTON VALVE.

Cast iron packing rings are fitted to the valve and constitute the edges of the valve. They are prevented from entering the steam-ports when the valve is in motion by the narrow bridge across the steam-ports of the bushing, as shown in Fig. 69. The operation of the valve is clearly shown in Fig. 68, the direction of the steam being indicated by arrows.

In setting these piston valves, only the high-pressure ports are to be considered. Both heads of the steam chest are removed, and with a tram, from some point on the body of the cylinder to the valve stem, the line and line positions of the valve in both front and back motion, are laid off and indicated by a prick punch mark on the valve stem. Using the same tram, the position of the

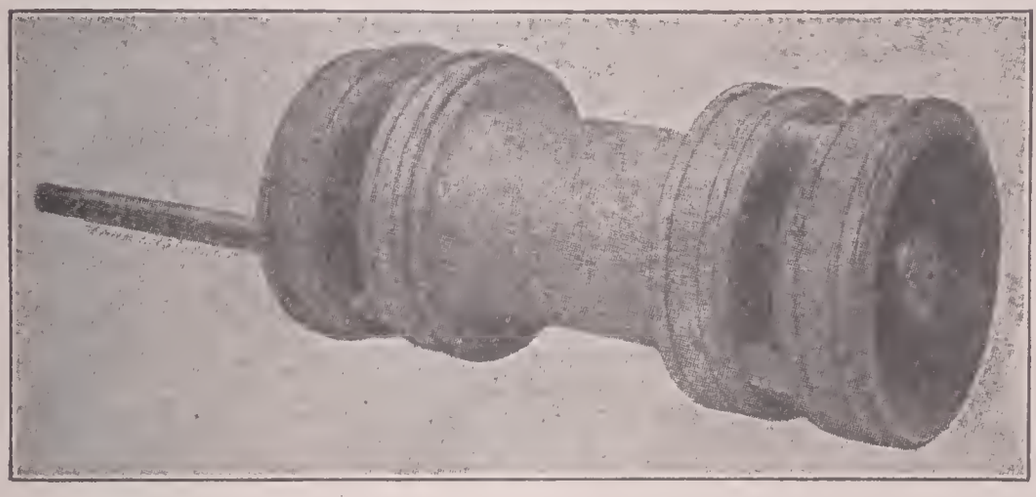


FIGURE 70. PISTON VALVE USED ON VAUCLAIN COMPOUND LOCOMOTIVE.

valve at different parts of the stroke can be ascertained, and the opening of the ports noted by the distance from the point of the tram to the prick punch mark. The relation of the low pressure ports to the valve must be ascertained by measurement, the same as the exhaust ports in ordinary slide valves.

THE AMERICAN BALANCE VALVE.

This form of balance can be applied to almost every form of slide valve.

The American balance valve Fig. 71, has been adopted by a great many of the principal railroads in this country,

consequently we give details of its construction, believing it will be interesting to a large number of our readers. It has also attracted the attention of foreign builders and is now in use upon many locomotives in other countries. It is claimed many thousand of these valves are in use, being used by 110 different railroads in this country and many foreign roads; also on marine and stationary engines. One of our illustrations shows its application to

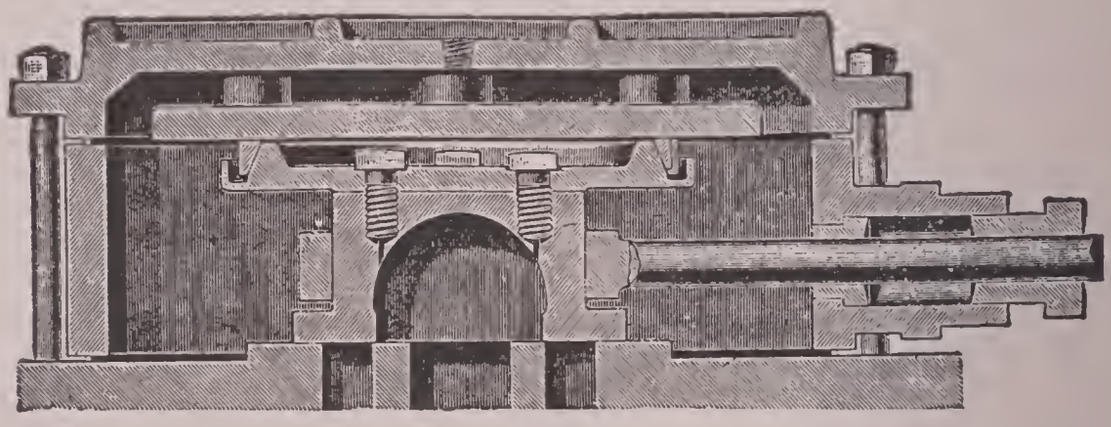


Fig. 71.

an Allen ported valve. The claims made for this form of balance are:

1. *Self supporting*—when not under steam.
2. *Supported by steam*—when under steam.
3. *Automatic adjustment*—with or without steam.
4. *Absolute steam joints*—no waste from leakage at any time.
5. *Positive action*—impossible for ring to stick.
6. *No lateral wear*—ring moves as part of the cone itself.
7. *Permanency of cone*—cones retain original dia. and taper.
8. *Standard sizes*—rings interchangeable, old or new.

9. *Stock*—carried in stock for new work or repairs.
10. *Lathe work*—stock from the lathe is economy.
11. *Greater area of balance*—equaled by no other design.
12. *Simplicity*—always most desirable in machinery.

It is claimed by the makers of this valve, (The American Balance Valve Co. of Jersey Shore, Pa.) that, first of all it furnishes an absolutely steam tight joint, not only when newly fitted, but all the time.

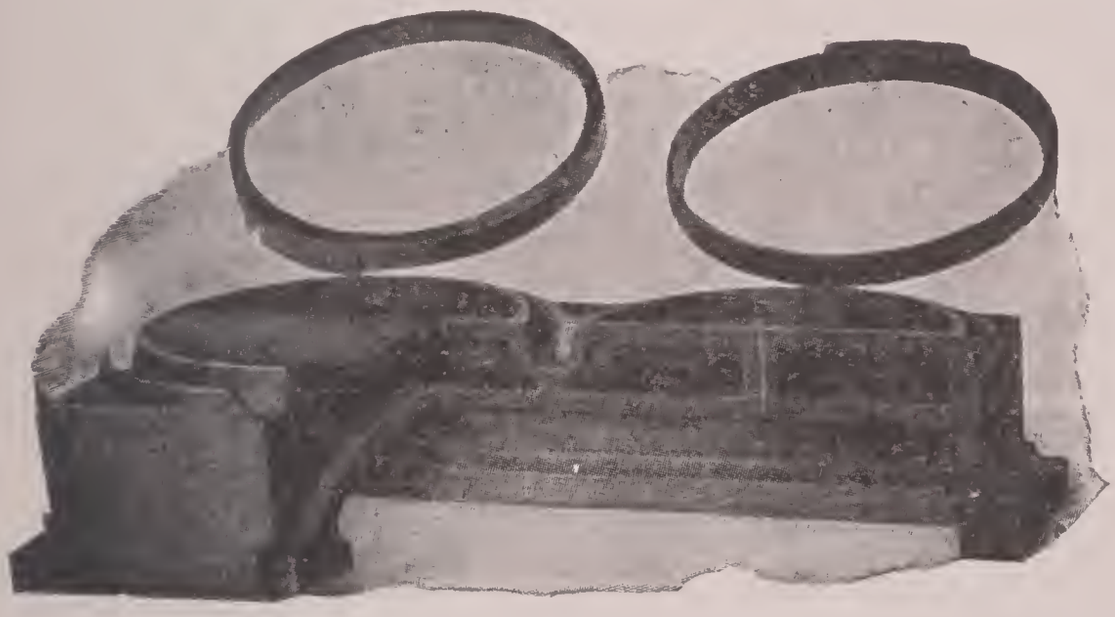


FIGURE 72. RINGS AND CONES—AMERICAN BALANCE VALVE.

Second, greater area of balance, the valve being balanced in what is presumably its heaviest position, and with the steam pressure acting upon the conference of the taper rings, (See Fig. 72) it will be observed that for the valve to lift off its seat, it is necessary to force the cone up into this taper ring, and since the ring is held by the steam pressure from opening, the valve cannot lift without first overcoming the entire friction of the beveled face, besides opening the ring against the pressure in the steam chest.

It will therefore be seen that whether the chest pressure be heavy or light the valve is always retained upon its seat.

It might at first glance appear to be a natural conclusion that this taper of rings and cones would crowd the valve down on its seat unnecessarily hard, and no doubt it would have that tendency if the degree of taper was made great enough, say 45 degrees for instance, in which case the action of the steam chest pressure upon the circumference of the ring would of course wedge it in between the cone and the chest cover, and thus exert an enormous pressure upon the valve.

Experience has however demonstrated that there is a correct degree of taper, which varies all the way from 9 to 24 degrees, to be given to the rings and cones, depending upon the style of valve, whether it be single "disc," double "disc," single cone, etc., and the claim is made that rings have run 190,000 miles with but 1-32 inch wear off their faces.

It is also claimed in favor of this system of balance, that it is positively automatic, in adjustment, also self supporting, with no delicate parts, and therefore not easily broken, having no springs.

The only repairs necessary on the American balance is to put in a new ring when the old one has worn out from the top downward.

The new rings are one inch in depth, and they can easily wear ⅜ inch, and still adjust themselves.

Assuming that the rings are made of the proper quality of metal, and that they receive the proper care in service, the time required to wear ⅜ inch from the face, would be from four to eight years of continuous service.

It will thus be seen that the cost of repairs is very

light, and another great advantage is, that when an old ring is replaced by a new one there is absolutely no change in the balance.

This is explained by the fact that since the steam pressure on the circumference of the ring holds it firmly against the beveled face of the disc or cone, while in operation, under steam (its own tension holding it when not under steam), there is absolutely no lateral wear on either the ring, or the risc, consequently a new ring fits an old disc.

FIGURE 73. T RING—AMERICAN BALANCE VALVE.

The rings are made from standard gauges which are used on the lathe in place of a caliper, or rule, and as the rings are all lathe work the time required for fastening on the L-shaped piece for covering the cut of the ring is very short, not more than twenty minutes hand work.

In turning up the rings, they are made ¼ inch smaller in diameter than their working diameters. They are then cut, and expanded over the cone which gives them a tension that makes them self supporting when not under steam.

The pressure of the steam on the circumference of the ring supports it when in operation.

The latest improvement in this form of balance is the T ring (Fig. 73); the flanges on the top of the ring give an extra width for wearing surface, which prolongs the life of the ring. The top face of the ring is provided with a small groove for oil. The L-shaped joint plate forms joints both on the beveled face and at the top of the ring. The "outside rim" or flange extending outside the taper ring is to prevent pieces of the ring from falling in the path of the valve in the event of accident to the ring.

SINGLE "DISC" BALANCED VALVE.

Always use the single balance where chest room will permit it, as one ring and disc is simpler than two.

For length of steam chest for single balance add the extreme travel of the valve to the outside diameter of disc, and to this sum add not less than ½ inch for clearance—¼ inch at each end of chest. If a little more clearance is desired the rims of disc may be cut ⅛ inch—just flattened on two sides in line of valve travel. But in no case are they to be cut beyond their inside diameter.

If sufficient clearance cannot be obtained by cutting the rims ⅛ inch each side in line of valve travel, then double balance must be used.

DOUBLE "DISC" BALANCED VALVE.

When the steam chest is too short to leave clearance for the outside diameter of the disc, or cone of single balance at extreme travel of valve, it then becomes necessary to use double balance.

If the yoke fit (or box) of the valve is large enough two cones are cast on the valve, but in case the yoke fit is not large enough to cast cones on, then two discs are used.

If the distance across the two discs when they are side by side on top of the valve is greater than the width of the steam chest, the rims on each disc may be cut ⅛ inch at the center of the valve, thus drawing the discs ¼ inch closer together, and if more clearance is still required the rims may also be cut ⅛ inch at the ends of the valve, thus giving ¼ inch more, making a total of ½ inch.

But in no case should the rims be cut more than ⅛ inch or to their inside diameter.

If discs thus cut will not clear the sides of the steam chest, it will be necessary to use a smaller balance.

"DISC" BEARING ON CONE.

In all cases where possible the height adjustment should be made by lowering the cover of the steam chest, or bearing plate, but when this cannot be done, the discs may be raised.

When it becomes necessary to raise the disc on the valve, longer bolts should be used, and the liners placed between the disc and the valve must be true, and large enough to give a solid bearing for the disc on the valve.

If it is found necessary to raise the disc to clear the top of the valve, the same rules must be observed.

The bolts that hold the disc to the valve should be steam tight on threads, also steam tight under the heads, a copper washer being used under the heads, thus forming a bolt lock.

The interior of each disc or cone is relieved to the exhaust cavity of the valve.

In "cone" balance, relief holes are drilled through the top of the valve, but in "disc" balance these relief holes (¼ inch in diameter) pass through the bolts, one hole through each bolt.

SINGLE "CONE" BALANCED VALVE.

This style of valve must be cast flangeless if a valve yoke extending all around the valve (as in locomotives) is used, but need not be flangeless when made for center rod to drive the valve (as in stationary engines). In case of the locomotive yoke we recommend the yoke to be carried on the steam chest at the ends of the valve. Where old chests have rubbing strips wide enough they can be planed on top and the yoke allowed to ride on them, and in new work this can be done cheaper than to put on a front carrying horn, and is more efficient than to support the yoke on the valve stem packing the valve itself. A valve need not be flangeless to thus support the yoke, it can be carried with any valve, and it insures the free upward movement of the valve at all times, which is very essential in obtaining the best results.

OUTSIDE RIMS.

The outside rim on disc or cone is merely a safeguard to the ring in case of accident—it performs no other duty. The required inside diameter of this rim must allow the ring to be expanded on the cone until the top face of the ring is flush with top of cone and still clear the ⅛ inch joint plate on the outside of ring. In single

balance the rims may be cut ⅛ inch front and back, giving ¼ inch more clearance, when the disc runs too close to steam chest at full travel of valve.

In double balance the rim of each disc may be cut ⅛ inch so as to draw the discs ¼ inch closer together at center of valve, and if more clearance at sides of chest is required the rims of each disc may be cut at ends of valve also. The distance across the two discs can thus be shortened ½ inch.

In no case shall the rim be cut more than ⅛ inch. The two cones where they come together at center of valve in double balance must not be less than ¾ inch apart at the bottom.

If discs thus cut will not clear the steam chest, then smaller balance must be used.

HEIGHT ADJUSTMENT.

When the valve is in position and the chest cover has been screwed down there must be ⅛ inch between the face of the bearing plate (sometimes called balance plate) and the top of disc or cone.

The rings are bored for this position and in this position have their proper tension. This allows the valve to lift off its seat ⅛ inch, which it will do as soon as steam is shut off while the engine is in motion, provided it is not held down by the valve yoke.

The valve yoke must not interfere with this upward movement of the valve.

Rings are all bored smaller than the diameter at which they are to work; therefore when a ring is set on its proper cone it will stand higher than its working position.

The face of bearing plate must not be closer than ⅛

inch to top of cone after chest cover has been screwed down. In placing the cover in this position the ring is expanded over the cone until its inside diameter at bottom is the proper balancing diameter.

Owing to the natural elasticity of the ring and its expansion over the cone, a tension is placed on the ring, the action of which is (the same as the steam pressure) to close the ring on the cone, which necessarily moves upwards.

The ring is therefore self-supporting and self-adjusting. All rings are interchangeable on discs and cones of respective sizes whether standard or special.

BALANCES.

American balances are known under the following heads, one valve being balanced in each case:

Single "disc" balance, i. e., one ring and one disc.

Double "disc" balance, i. e., two rings and two discs.

Single "cone" balance, i. e., one ring with cone cast on valve.

Double "cone" balance, i. e., two rings with two cones cast on valve.

CYLINDER RELIEF.

The valve shall always be free to lift ⅛ inch off its seat, to allow the free passage of air from one end of the cylinder to the other, between valve and valve seat, when engine is running without steam. The tops of all American balance discs, or cones, show a polish, giving positive evidence of their contact with the bearing plate or cover, and that they therefore do float.

The explanation is: At the first stroke of the piston, after engine has been shut off, air is compressed in one end of the cylinder while the valve is traveling a distance equal to its outside lap; at an early stage of this compression the valve is thrown off its seat and the escaping air rushes under the valve into the opposite end of the cylinder to relieve the suction which is taking place in that end; this operation is repeated so rapidly that the valve is kept floating until a slow speed has been reached.

Sufficient air is always drawn in, through the (lift valve) cylinder cocks, and exhausted through the exhaust port, to keep the current of air in the direction of the stack, but not enough to "fan the fire." This affords the most perfect cylinder relief and vacuum valves in steam chest are not a necessity.

FORMULAE OF BALANCE USED (AMERICAN BALANCE VALVE).

Area of balance for plain valves.

Area of one steam port, two bridges, and the exhaust port, plus 8% if for single balance, and plus 15% if double balance.

Example for single balance.

Steam port, 1¼" plus bridge 1" plus exhaust port 2½" plus bridge 1" equals 5¾" × 16" equals 92 sq. in.
8% of 92 equals 7.36.
92 plus 7.36 equals 99.36 sq. in. equals area.

Example for double balance.

Steam port 1¼" plus bridge 1" plus exhaust port 2½" plus bridge 1" equals 5¾" × 16" equals 92 sq. in.
15% of 92 equals 13.80.
92 plus 13.80 equals 105.8 sq. in. equals area.
105.8 divided by 2 equals 52.9 equals area of each ring.

The nearest diameter for 52.9 equals 8¼ in., which is the diameter for each ring.

For Allen ported valves, use the same formulæ as above, then from the area derived, subtract the area of one side of the Allen port.

Example.

1½ plus 1⅜ plus 3 equals 7¼ in. × 17 equals 123.25 sq. in. plus 15% equals 141.73 sq. in. equals area.

Allen port equals ½ × 17 equals 8½ sq. in., then

141.73 minus 8.5 equals 133.23 sq. in., which divided by 2 equals 66.61 sq. in. equals area of each ring.

THE BARNES BALANCED VALVE.

The following description and illustration of the Barnes balanced valve is taken from *"Locomotive Engineering"*: A balanced valve having some points with a strong flavor of originality is that designed and patented by Superintendent of Motive Power Barnes, of the Wabash Railroad, and illustrated in Fig. 74.

The full size part of a longitudinal section will perhaps give a clearer idea of the balanced part of the device than will a first reference to details.

This view shows a cast iron frame surrounding the valve, with a recess equal to the width and depth of the balance strips on its inner face.

There are four balance strips ⅝x1¾ in. carried in position by the frame, which latter is 1⅝x2½ in. in the overall dimensions of its cross section.

Both the frame and the strips are cored for lightness.

The combination is held up to the balance plate by four helical springs made of German silver wire 0.125

in. in diameter. It is seen that the balance strips have a very deep bearing between that valve and frame, and since the strips are carried by the frame, there must be a constant depth of support to them on the outside face, no matter how much wear takes place, nor what the lift of the springs. This condition of things tends to reduce the liability of cocked or broken strips as has been found in service. The springs placed at the ends of the frame

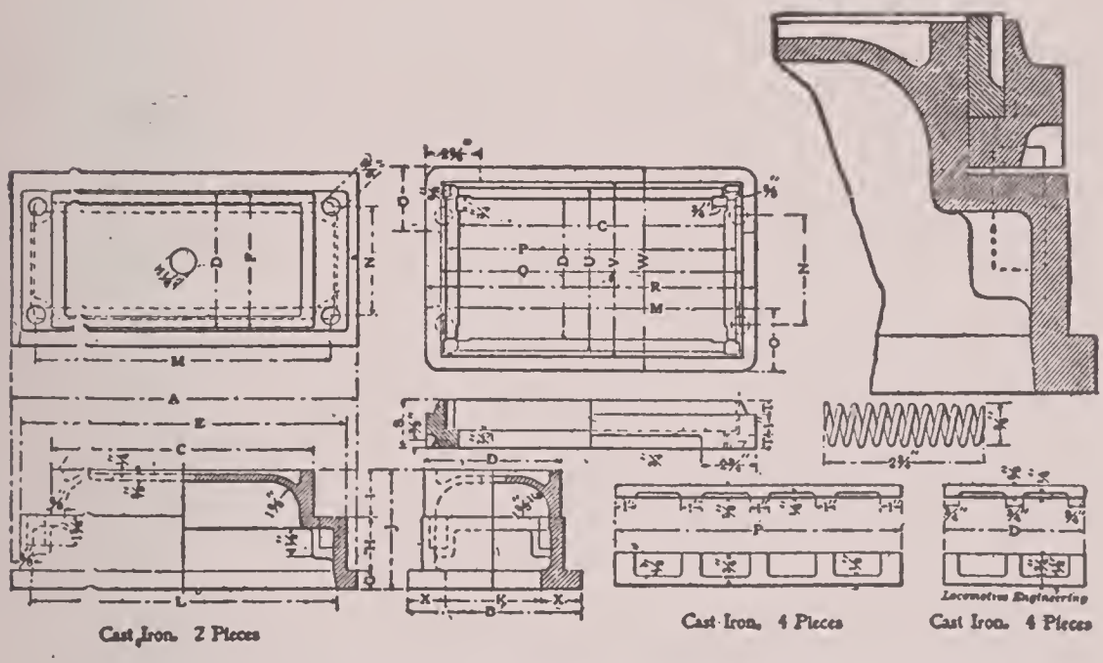


FIGURE 74. BARNES BALANCED VALVE—LONGITUDINAL SECTION.

would seem to exert a more equable pressure on all the strips than is possible with a long spring under each individual strip, for the reason that it is a delicate undertaking to attempt to make four flat springs of the same degree of elasticity.

On these grounds it is not unreasonable to expect a continuance of the good results so far shown by this style of balance. The dimensions shown are for a ten wheeler.

THE McDONALD VALVE.

The points of advantage claimed for this valve are as follows: First, that the pet cock on top of the steam chest cover is always open to the atmosphere, so that should the joint leak, the engineer will be able to see the steam escaping through the pet cock, which is sure evidence that the valve requires attention.

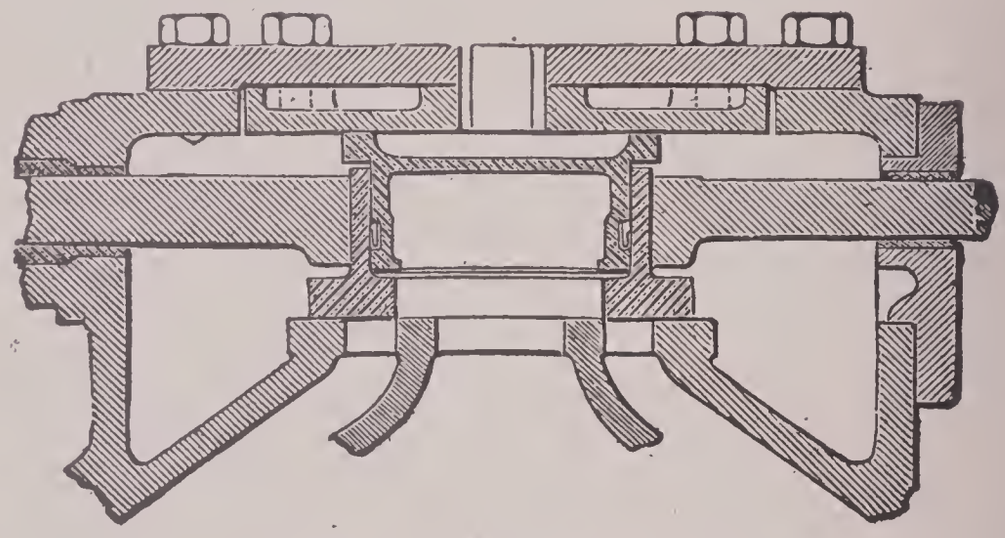


FIGURE. 75. McDONALD VALVE.

Second, he can then close the pet cock and the valve will operate on the same principle as a simple D valve.

Third, the U shaped packing strips maintain a satisfactory and tight joint and the flat steel spring beneath the valve is of the simplest form. This valve has recently been introduced into this country. Mr. McDonald, the inventor, is a mechanical engineer at Yokohama, Japan, and has had these valves in service some two or three years, and it is claimed they are giving excellent results.

The McDonald valve is shown in vertical section in Fig. 75.

THE BRIGGS BALANCED VALVE.

The following illustrations (Figures 76 to 80) and description of this valve is taken from *Locomotive Engineering:*

The inventor, Mr. R. H. Briggs, Jr., of Amory, Miss.. describes this valve as follows:

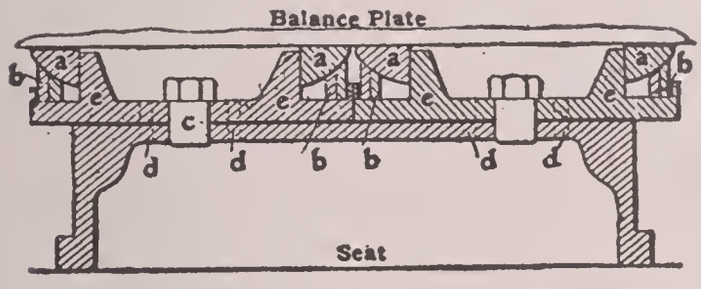
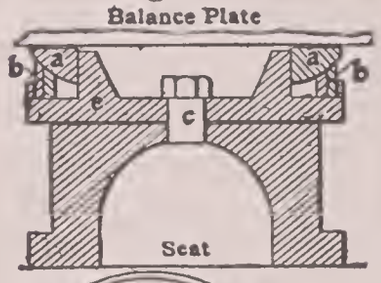


Fig. 76.

Fig. 77.



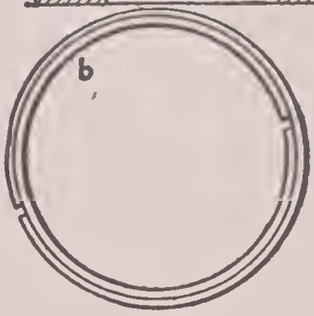


Fig. 79.

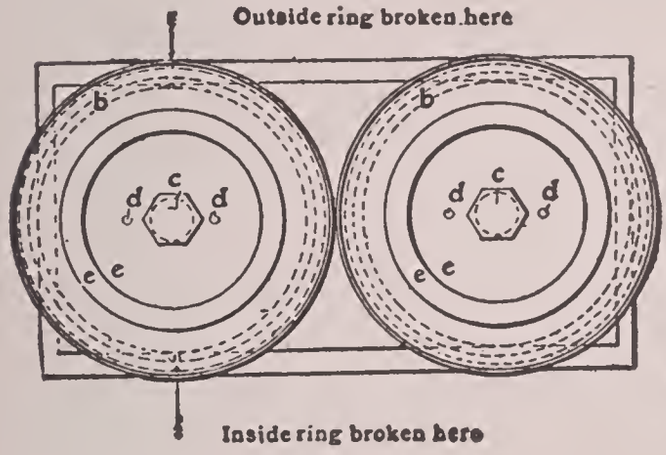


Fig. 78.

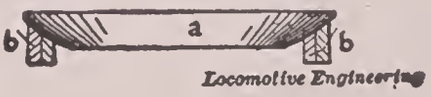


Fig. 80.

"You can readily see that it will require very little explanation as to the operation of this improved balance. This balance consists of four parts—cone E, two packing rings B and joint ring A. The packing rings are cut as shown in Fig. 79. To break the joints these rings are pinned together. To prevent the cuts working around,

they are also ground on cone E, to prevent any leak at bottom of packing ring. They have a ball joint on top with joint ring A perfectly ground to them. Joint ring A is a nice, neat fit on cone E. Now you can very readily see that it is impossible for this valve to blow, as the greater the steam pressure on the outside of packing rings B the tighter ring A will be on the balance plate. I give these rings only 1-32" compression. Another advantage that I claim for this valve is that it balances to outside diameter of outside packing rings B."

THE MARGO VALVE.

The Margo balanced valve was one of the first forms of balanced valves. It had two small discs and small packing rings (similar to air pump packing rings); the steam pressure holding the disc up against the pressure plate. Much trouble was experienced with this form of valve, owing to a gumming up of its parts which would cause the disc to stick. Very few of these valves are now in use.

POWER REQUIRED TO MOVE A VALVE.

To determine the power required to move a valve multiply the area of the valve face by the steam pressure upon it, less ⅓ allowed for back pressure from the steam port and exhaust port. The friction between two smooth surfaces well lubricated varies from 1-10 to 1-14 of the pressure (the weight of the valve being so slight it is seldom considered. Friction is the resistance which two contracting surfaces have to being moved one over the other and is of three kinds: Sliding, rotation and liquid). For ex-

ample: If a valve face measures 10x20 with 120 pounds' pressure proceed as follows:

$10 \times 20 = 200 \times 120 = \ldots\ldots\ldots 24{,}000$

$8{,}000$ less one-third,

Divided by the friction, 10..... 16,000 1,600 lbs. power required.

THE ALFREE VALVE.

In the Alfree system of steam distribution as applied to locomotives by the Locomotive Appliance Co. of Chicago, a somewhat new departure from standard practice is made.

This consists in what is termed a compression controlling valve, in addition to the main valve which controls admission, cut-off, and release. According to the claims of the manufacturers, "this system not only embodies principles that are entirely new, but utilizes old and well known principles in the use of steam, not heretofore suitable for locomotive uses because of the complication of the mechanical means necessary for their use, together with the peculiar and severe conditions under which they must operate.

"The mechanical means for producing these results are extremely simple; the few special features being contained within the valve chambers of the cylinders. All that is necessary for an application of the system to a locomotive is to apply our cylinders in precisely the same manner as standard cylinders, connecting the valve stems to the standard rockers and making the usual adjustments. It involves substantially a change of cylinders only. Any of the standard valve gears, Stephenson, Walschaert or any other may be used.

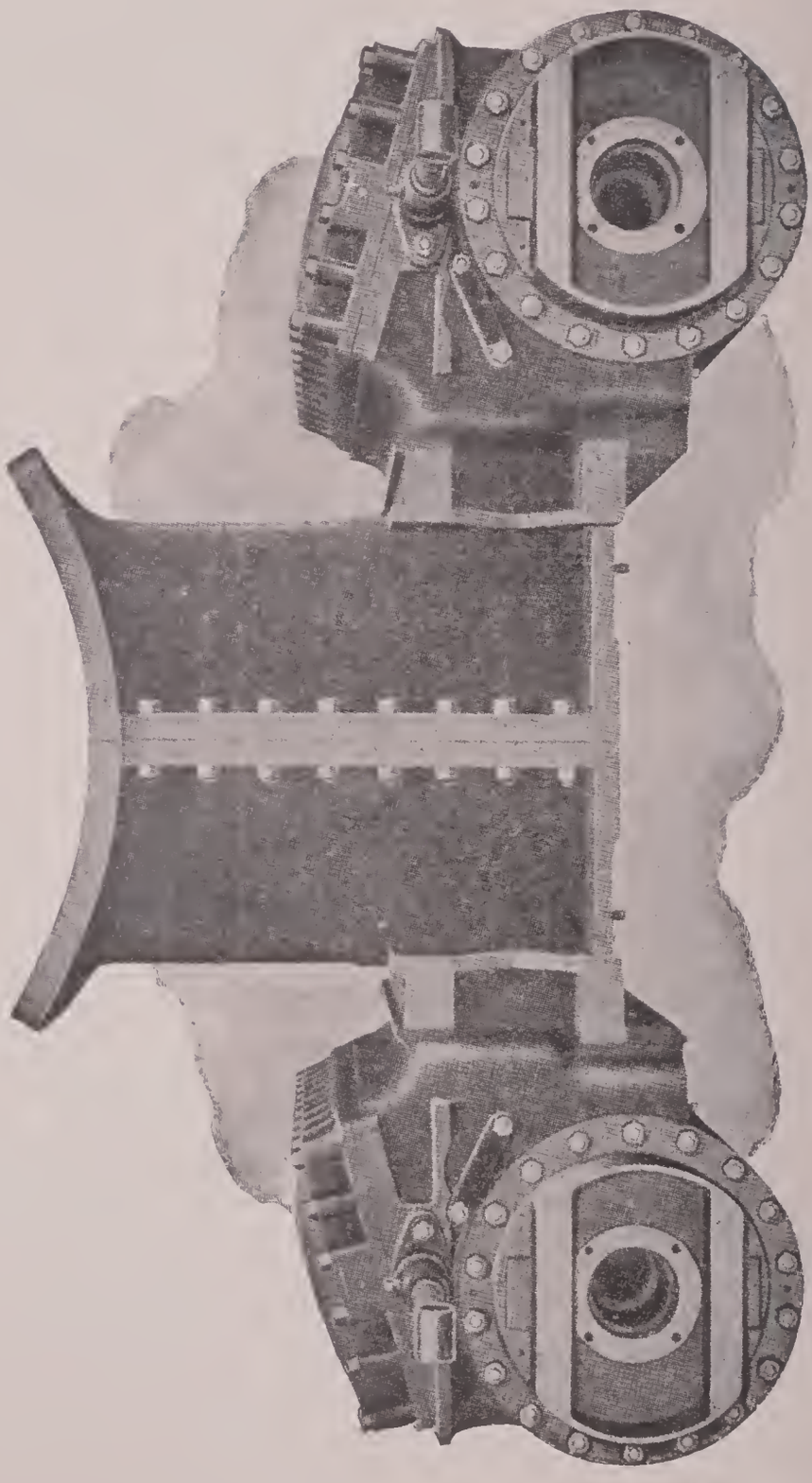


FIGURE 81. END ELEVATION OF ALFREE VALVES AND CYLINDERS.

"The following is a brief analysis of the means through which these results are attained:

"1. *Reduced Heat Losses.* Our cylinders are designed with short ports and the exhaust passages carrying cold steam are separated and insulated from the live steam passages; consequently heat losses from radiation and condensation are greatly reduced.

"2. *Less Steam Used for a Given Cut-off.* The waste spaces in the cylinders are reduced to the minimum. In common practice this amounts to at least 10% of the piston displacement, while our cylinders have not more than 2.5%. To illustrate, for a 7-inch cut-off in a 30-inch stroke engine, we, in effect, draw upon the boiler for 7¾ inches of steam, while for a like cut-off in a standard engine it will require an average draft on the boiler of 10 inches of steam.

"3. *Higher Ratio of Expansion.* Because of the minimum waste spaces in the cylinders and of the ability to hold the steam in the cylinders longer, before exhausting it, we get a higher ratio of expansion—that is, more work—out of the steam admitted to the cylinders.

"4. *Perfect Exhaust.* Having an increased exhaust area of about 50%, this system produces a quick, extraordinarily free, and large opening of exhaust, discharging the exhaust steam perfectly and quickly. This greatly reduces back pressure, or negative work, in the cylinders, and, due to the sudden release, a better fire is maintained even with an enlarged exhaust nozzle.

"5. *Reduced Negative Work.* In the return or exhaust stroke of the piston, the exhaust is not closed until very late in the stroke; but when the proper point is reached, just enough exhaust steam is entrapped in the cylinders and small clearance spaces to produce sufficient

compression to answer the mechanical needs of cushioning. In the standard locomotive the exhaust is closed comparatively early in the stroke, and negative work commences correspondingly early. As the compression begins early in the standard engine, a large amount of clearance space must necessarily be provided to prevent compression running too high, and a correspondingly large amount of negative work must be done to compress the larger volume. Therefore, the negative work of compression in our engines is reduced enormously, and a corresponding amount of available positive work is added.

"6. *Increased Turning Effort.* It is found in locomotives where the power is applied to the drivers through rotative rather than reciprocating motion that such locomotives pull, without slipping of the drivers, considerably greater tonnage than the standard locomotive of the same weight. The reason for this is that the power is applied steadily in one case, and intermittingly in the other. In the reciprocating engine, if the release is early and an unnecessary resistance set up on the opposite side of the piston, more steam must be applied in the center of the stroke to make up for losses at each end. Consequently, the power is applied to the drivers intermittingly, producing a jerky effect, and the limit of adhesion of the drivers is reached long before the real power of the engine can be used. But in our system of holding onto the steam longer at one end, and by removing the resistance at the other end of the stroke, higher average pressure will result and the power distributed more evenly through the stroke; thus imparting to the drivers a steadier or more even rotative effect, increasing the available power of the engine and greatly decreasing tendency of slipping.

"It is a well established fact in standard locomotive

practice, that a single valve either of the piston or flat type, will perform three out of its four functions perfectly, viz.: admission cut-off, and release, but the fourth, known as closure, or the compression event, is wrong. If the valve is provided with a sufficient amount of exhaust lap, to obtain an economic expansion through delaying the release, the exhaust closure becomes excessive early, and an enormous clearance space must be provided equal to fully 15% of the piston displacement. This entails such a loss that a compromise is generally effected by reducing the exhaust lap, making the release earlier, but delaying the closure to a point where not to exceed 10% of clearance space need be provided. This requires a valve having its exhaust edges about equal to the exhaust edges of the port, generally designated as line and line, or if the valve is cut a little shorter, giving say about one-eighth exhaust clearance, the clearance or waste spaces may be reduced to about 8% of the piston displacement. Any further reduction of exhaust lap would result in a greater loss through an early release, than would be gained through a reduction of clearance.

"In the Alfree System a single valve correctly controls admission, cut-off and release the same as in the standard engine, except that a sufficient amount of exhaust lap is used to carry the steam to a point that will give a greater expansion. Then to avoid an early closure, a small piston valve, called the Compression Controlling Valve, is introduced through a section of the ports beneath and to one side of the main valve, and has only the function of controlling the compression and providing greater freedom for the escape of exhaust steam. While the two valves release at the same instant, the compression valve in closing falls about one and one-fourth inches behind the main

valve. This allows the exhaust steam that would otherwise be in compression, to escape until the piston reaches about 90% of its stroke (at ¼ cut-off) or within 2½

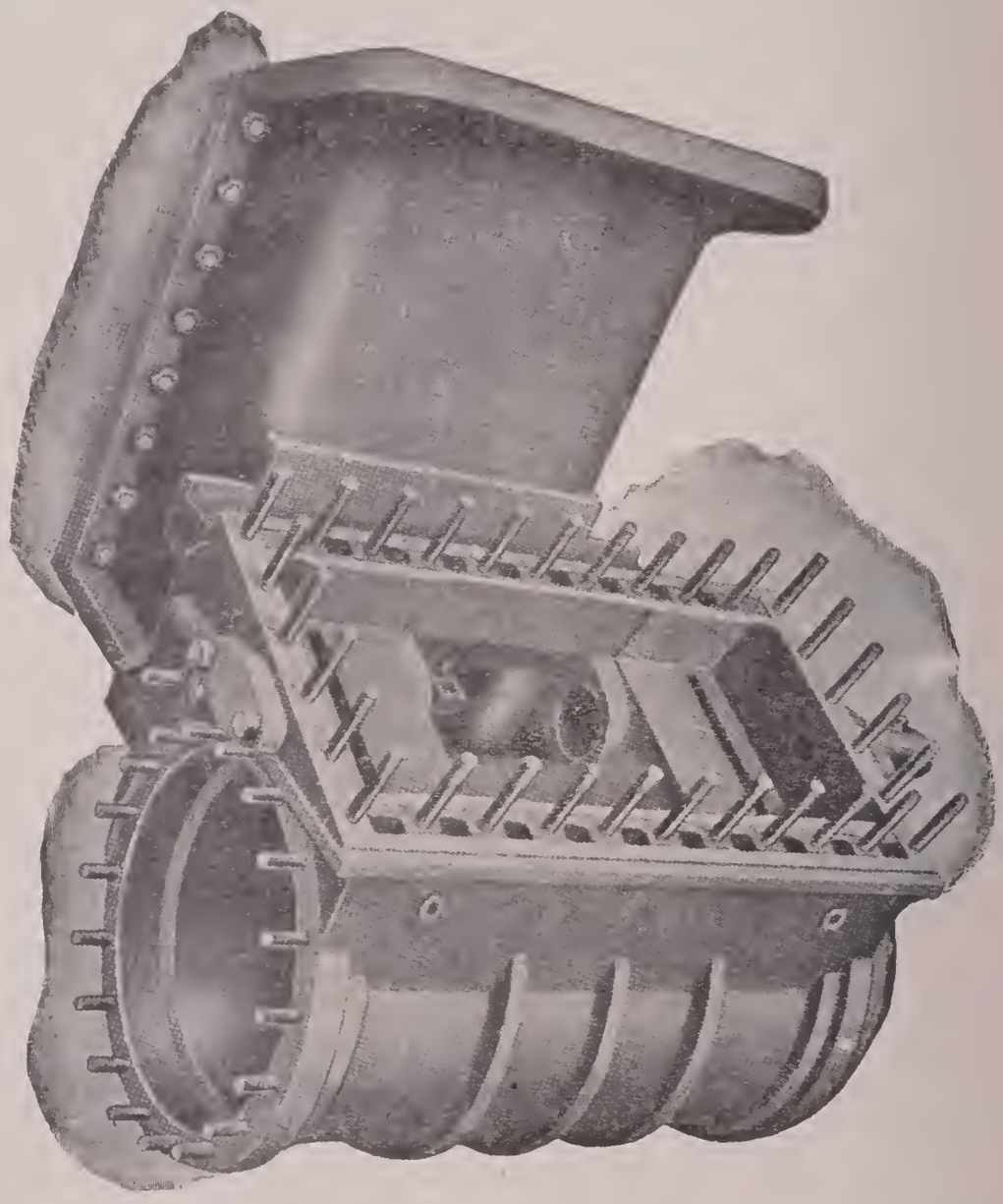


FIGURE 82. ALFREE CYLINDER, WITH STEAM CHEST COVER, AND VALVES REMOVED.

or 3 inches of the end of its stroke. The clearance having been reduced to 2½% of the piston displacement, a sufficient amount of compression takes place to perfectly cushion the reciprocating parts.

"It will be seen that through these simple means we are enabled to greatly reduce the inherent losses, increase the mean effective pressure, increase the effective exhaust area, and increase the turning effort on the drivers. The results as found by actual test are 5% to 8% more tonnage, with 8% to 10% less fuel and water, higher speeds, fewer engine failures, and reduced running repairs. These are positive and practical facts which will not only be found true by critical examination of the system, but will be better proven by actual use."

Fig. 82 shows the Cylinder with Steam Chest Cover and Valves removed. As will be seen, the Cylinder and

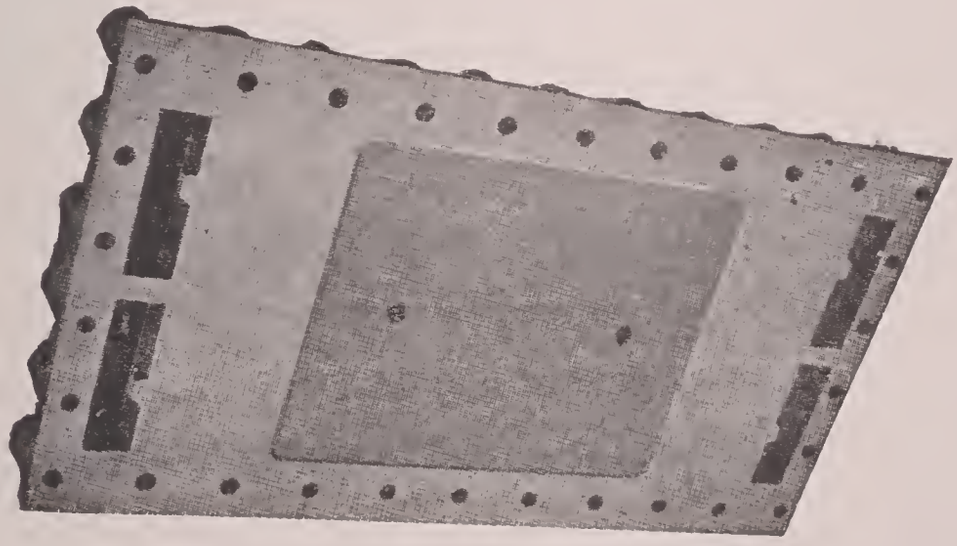


FIGURE 83. STEAM CHEST COVER.

Valve Chamber or Steam Chest are one casting; the ports are short and practically straight with the Compression Valves passing through their longest section. The admission is from the inside, which brings the live steam in contact with a considerable portion of the Cylinder walls, while the exhaust is at the ends and is most effectually insulated from the live steam, thus eliminating one of the

greatest sources of heat loss. Double walls are used around all live steam passages wherever practicable with the object of making the insulation as perfect as possible. This design produces an exceptionally strong cylinder well adapted to resisting shocks which would often destroy Cylinders of the ordinary design.

The Combined Steam Chest Cover and Pressure Plate, as shown by Fig. 83, has a by-pass chamber connecting with each end, thus permitting a portion of the exhaust from one end to flow through the exhaust passage at the opposite end, which greatly aids in preventing the induction of cinders into the cylinder when drifting. Between this by-pass chamber and the face of the cover, a dead air space is provided, insulating the live steam from the exhaust steam, while the exhaust steam, passing through the by-pass, insulates the dead air chamber from the atmosphere, which is somewhat further shut off by the usual cylinder casing. The cover also serves as a Pressure Plate for the Main Valve.

GENERAL ARRANGEMENT OF PARTS.

The general arrangement of Valves, Valve Chambers, Steam Chest Covers, etc., is shown in Figs. 84 and 85.

The Steam Chest or Main Valve Chamber is on an angle of 15° with the horizontal and placed close to the cylinder bore. The ports are short and practically straight so that their surfaces may be scraped clean and smooth. The Compression Controlling Valve Chamber passes through the longest section of the ports, and is bushed in the usual way to provide for repairs. The Steam Chest Cover forms the top side of the Main Valve Chamber and provides a by-pass connection between the exhaust passages which equalizes exhaust pressures.

The Main Steam Valve is of rectilinear form rigidly constructed and designed so that the wear is uniform regardless of the travel. It is balanced for all speeds and under all pressures, running or drifting, and with its wearing surfaces nearly doubled by special riding shoes, the wear on the Valve and Valve Seat is very slight.

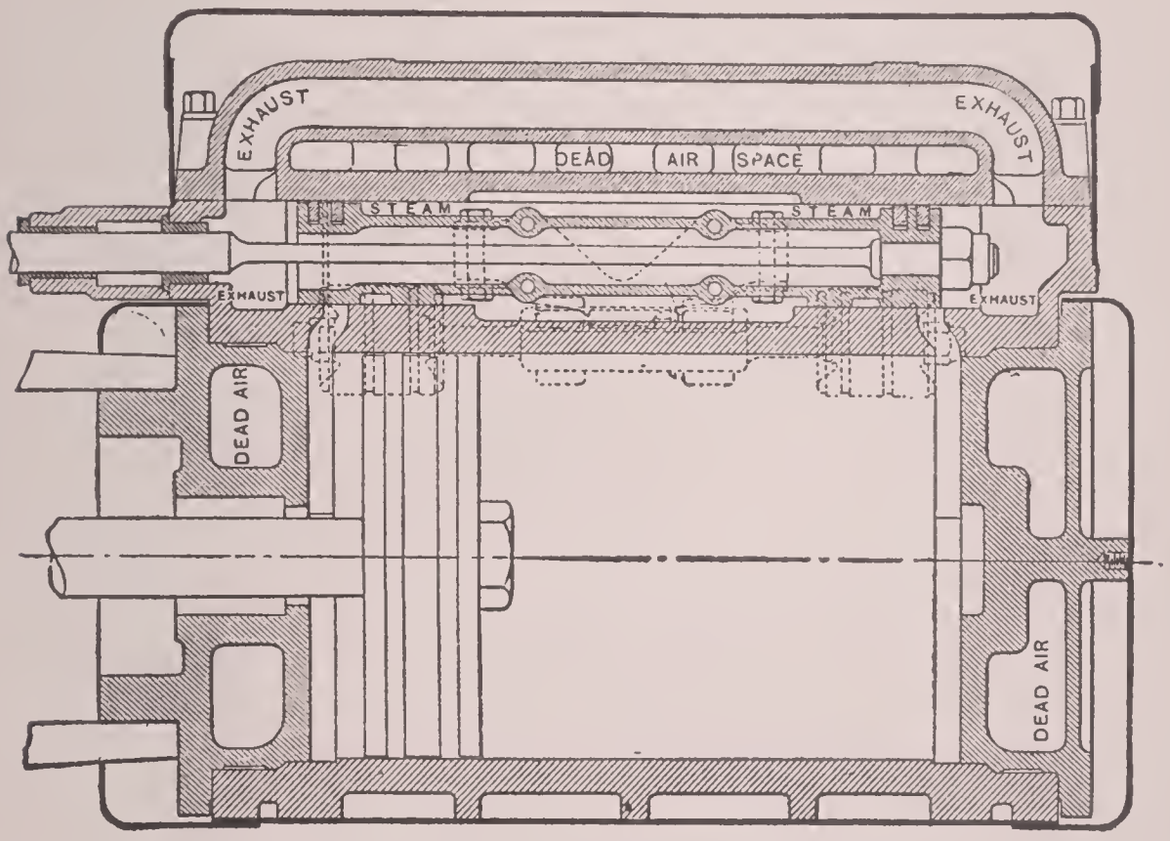


FIGURE 84. THE ALFREE SYSTEM.

The Compression Controlling Valve is a piston valve of the usual design except that it is very light and provided with wide snap rings to overlap the ports and for protection against wear. It takes its motion from the Main Valve by means of an arm connection of rigid construction.

The admission of steam to the cylinders and the cut-off is controlled by the Main Steam Valve. Steam is admitted along the entire lower edge and up one side, pro-

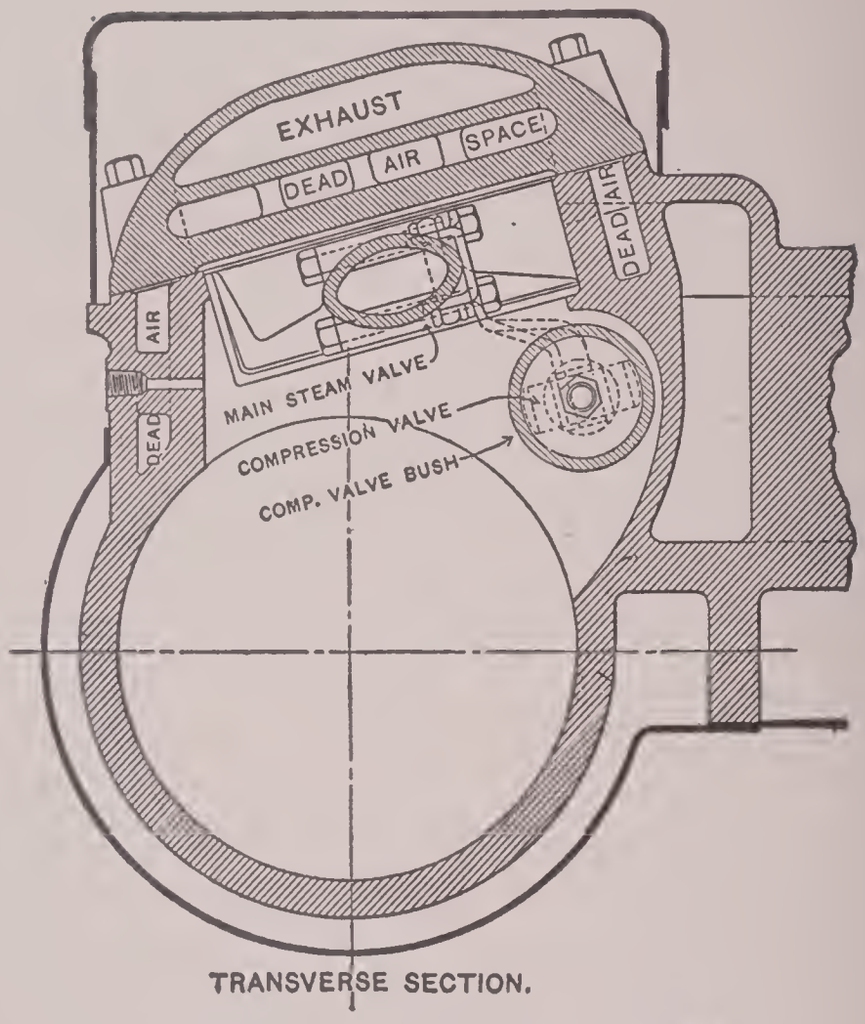


FIGURE 85. THE ALFREE SYSTEM.

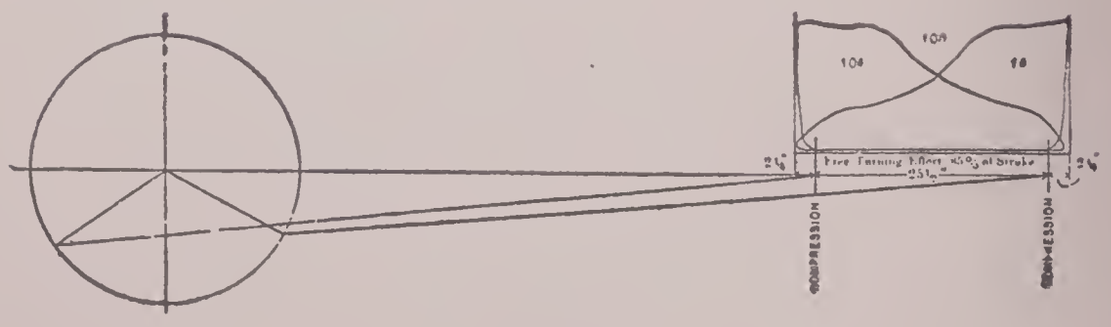


Fig. 86.

viding an unusually large port area for admission. Exhaust also is controlled by the Main Steam Valve, but to this is added the exhaust of the Compression Controlling Valve, which releases simultaneously with the Main Valve.

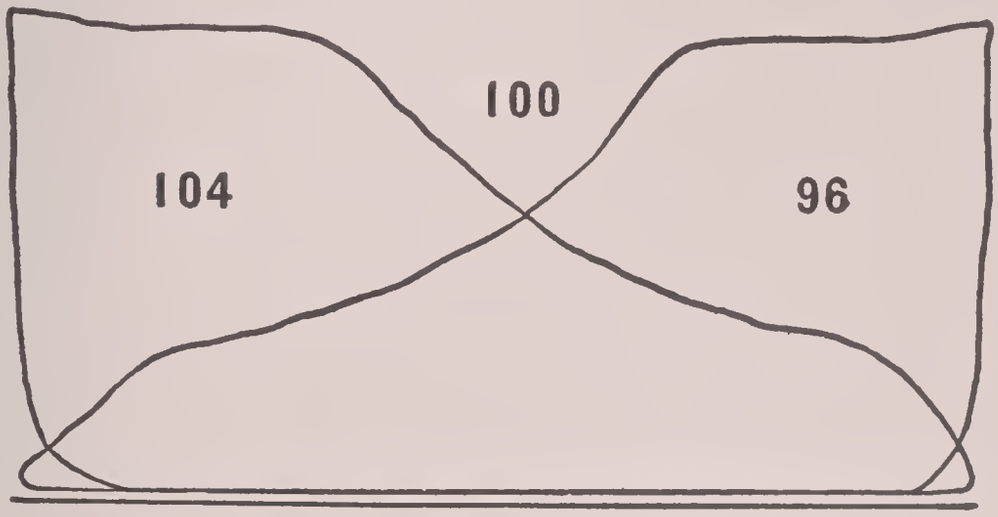


Fig. 87.

Closure or Compression is controlled entirely by the Compression Controlling Valve, which delays the final closure of the exhaust until the piston has completed 90%

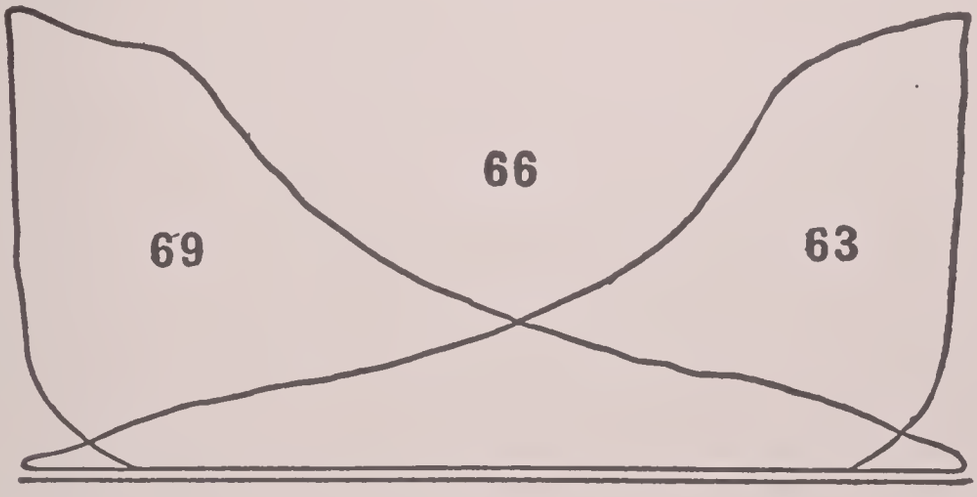


Fig. 88.

of its stroke at the short cut-offs, thereby reducing the volume of the exhaust steam in Compression from about 10 inches in the usual practice to 2¾ inches in this system.

Fig. 86 is a diagram furnished by the Locomotive Appliance Co. of Chicago, which that company claims shows the effective turning effort of the piston to be applied throughout the greater part of a revolution of the drivers.

Fig. 87 is an enlarged sample of the same card. Fig. 88 is another sample of an indicator diagram taken from a locomotive equipped with the Alfree cylinders and valves.

These diagrams certainly show great efficiency in the use of the steam.

Questions.

207. Are there any other types of valves used on locomotives, besides the D slide valve?

208. Mention a few of them.

209. What is the principal objection to the D slide valve?

210. Is the piston valve a perfectly balanced valve?

211. What pressure tends to press the D valve against its seat?

212. Is there any pressure to counteract this?

213. What are the causes of friction in piston valves?

214. Are all piston valves outside admission valves?

215. Why are piston valves practically balanced?

216. Why are piston valves made as long as possible?

217. What controls the admission of steam to the ports of a piston valve engine?

218. How is an outside admission piston valve set?

219. How is an inside admission valve set?

220. What advantage results from using an inside admission piston valve?

221. Name another advantage in inside admission?

222. What is one of the first duties of an engineer taking charge of a piston valve engine?

223. Why should he do this?

224. Are all engines equipped with indirect valve gear?

225. Repeat four simple rules for the guidance of an engineer in the study of valve gear.

226. Mention four possible combinations that may have to be dealt with.

227. In the first two of these, what are the positions of the eccentrics with relation to the crank pin?

228. What time would the hands of your watch indicate to represent this setting?

229. What would be the positions of the eccentrics relative to the crank pin in the third and fourth combinations?

230. What time would your watch indicate in order to correspond with this setting?

231. What is a good rule to remember in setting piston valves?

232. What type of piston valve does the American Balance Valve Co. manufacture?

233. Describe in general terms the construction of this valve.

234. What force expands the packing rings of this valve?

235. What prevents excessive expansion of these rings?

236. What is a common defect of snap ring piston valves?

237. How may the valve cage be worn unevenly?

238. What should a piston valve do in order to give efficient service?

239. What type of piston valve appears to be the favorite with builders?

240. Mention another advantage possessed by the piston valve over other forms of slide valves.

241. What does the term balanced valve include in its definition?

242. Have there been very many types of balanced valves tested on locomotives?

243. By whom is the Jack Wilson high pressure valve manufactured?

244. Give a short description of this valve.

245. Is it single acting or double acting?

246. What is the function of the balance plate?

247. What is the object of the pressure plate?

248. Is this valve balanced at all points of its travel?

249. What advantage is gained by having the valve make a full stroke across the seat at every revolution?

250. Describe the Allen-Richardson balanced valve.

251. How is the balance plate of the Richardson valve secured in place?

252. How is a steam-tight contact maintained between the upper surface of the valve and the balance plate?

253. How are these packing strips held in position?

254. Does the unbalanced Allen valve wear well?

255. What is the object of the internal passage in the Allen valve?

256. Describe briefly the Goodspeed valve.

257. What are the three principal objects sought to be obtained by the invention of this valve?

258. Describe in brief terms the action of this valve?

259. What is the essential novelty of the Goodspeed valve?

260. What objects are sought to be attained by the use of the Haley slide valve?

261. Describe Haley's slide valve.

262. Describe the operation of the device.

263. What principle is involved in the design of the Metzger slide valve?

264. What advantages are claimed by the Watkeys valve?

265. What is one of the greatest drawbacks to the efficiency of high speed locomotives?

266. What advantages are claimed for the Farrer improvement as applied to the ordinary slide valve?

267. How is it applied to an engine, and what is the principle of the device?

268. What can be said of the balanced slide valve?

269. How many styles of Gould balanced slide valves are there?

270. What methods are employed to balance these valves?

271. What is a noteworthy feature in connection with the action of this valve?

272. In case it becomes necessary to reverse the engine while under high speed, what will be the action of the Gould valve?

273. What is the function of the vacuum relief valve?

274. Describe the action of the pressure relief valve.

275. What great advantage is gained by its use?

276. In case combined pressure and vacuum relief valves are placed on the low pressure cylinder head, at what percentage of the boiler pressure should they be set?

277. At what should the high pressure cylinder head relief valve be set?

278. Is the piston valve a modern affair in locomotive practice?

279. Mention some of the early inventors and users of piston valves.

280. Upon what does the piston valve largely depend for its perfection of balance?

281. What are the requirements in the construction of the piston valve?

282. What advantage has the piston valve as regards steam clearance?

283. What type of valve is employed in the Vauclain compound locomotive?

284. Is this valve perfectly balanced?

285. What kind of packing rings are fitted to this valve?

286. Describe the method employed in setting these valves.

287. What can be said of the American balance valve?

288. Mention twelve points claimed to be in favor of this type of valve.

289. What is the first and most important claim put forth for this valve by its makers?

290. What is the second claim of importance?

291. Does the pressure in the steam chest affect in any way the valve being retained on its seat?

292. What form do the balancing rings assume in the American balance?

293. About what degree of taper should be given the rings and cones?

294. What are the ordinary repairs required on the American balance?

295. What length of time will they wear in service with proper care?

296. Is there any change in the balance when an old ring is replaced by a new one?

297. How is this explained?

298. What is the latest improvement in this form of balance?

299. Name the various heads by which the American balances are designated.

300. How much lift off its seat is the valve allowed when the engine is running without steam?

301. What occurs in the cylinder after the steam is shut off?

302. Describe the Barnes balanced valve.

303. What are the points of advantage claimed for the McDonald valve?

304. What constitutes the balance in the Briggs balanced valve?

305. Give the rule for determining the power required to move a valve.

306. Mention some of the claims put forth for the Alfree system of steam distribution.

307. Name six general headings under which certain claims for superiority for this system are presented.

308. Wherein does the Alfree system differ from ordinary practice with relation to compression?

309. Is the main steam valve in this system balanced?

310. Of what type is the compression controlling valve?

VALVE DIAGRAMS.

The action of the valve, and the relative positions of the crank pin, and eccentric during the stroke, may be graphically illustrated by means of valve diagrams.

There are several different kinds of diagrams employed for this purpose.

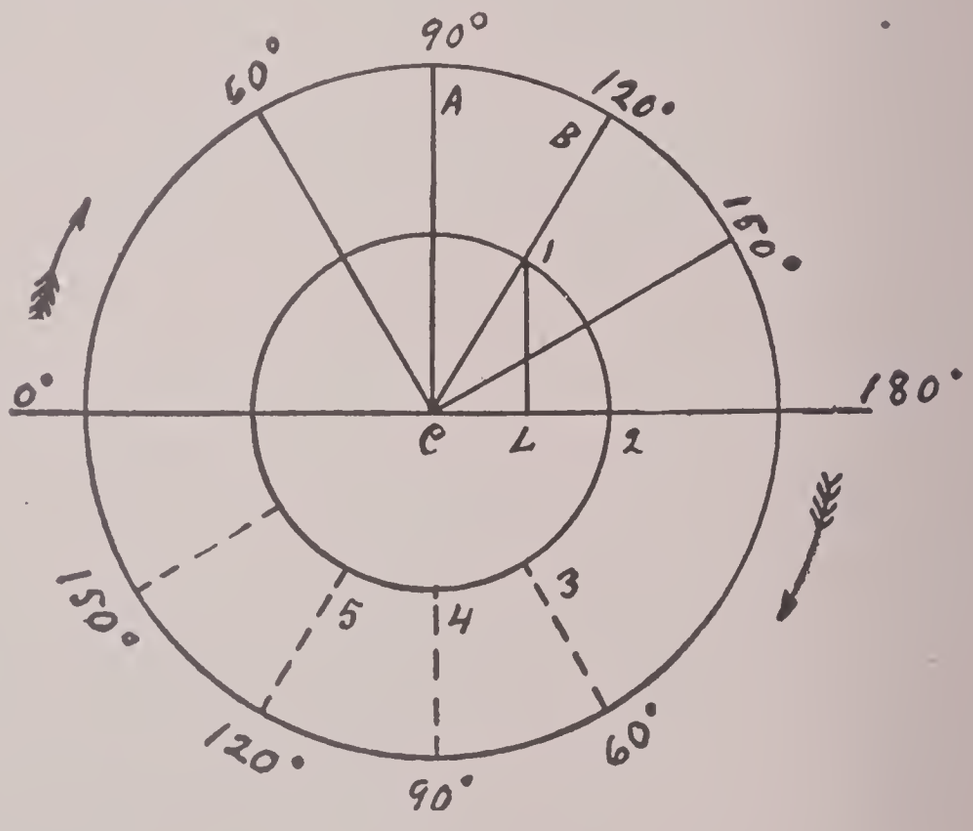


Fig. 89.

In this connection the Reuleax and Zeuner diagrams will be used, and first we will make use of a simple diagram in order to bring the subject clearly before the mind of the student.

Referring to Fig. 89, the inner circle represents the path described by the high point of the eccentric and the

large circle that of the crank pin. The radius C2 of the small circle represents the throw of the eccentric, and the distance CL is the lap of the valve plus the lead. The point of intersection of the vertical line, L1, with the eccentric circle locates the position of the highest point of the eccentric, and the line CB, drawn from the center of

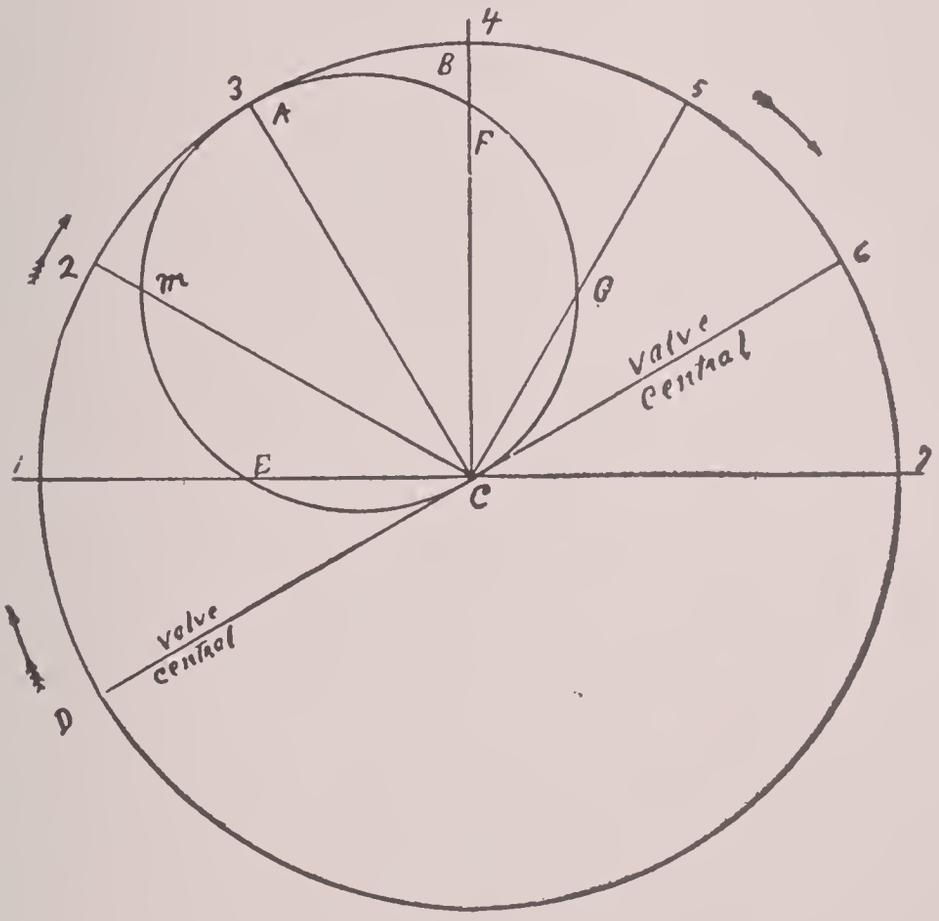


Fig. 90.

the crank shaft through this point, indicates the angular advance, which in this case is 30°, represented by the angle ABC. The figures 1, 2, 3, 4, 5 indicate the position of the high point of the eccentric at the moment of each function of the valve.

Fig. 90 shows the total movement of the valve, regardless of lap and lead. First draw line C1 to represent the center line of the engine. Next draw line C4 perpendicular to the line of centers, with C as the center of the crank shaft. The radius of the semi-circle D, 1, 2, 3, 4, 5, 6 equals the radius of eccentricity. Line CD represents the position of the crank when the valve is at mid-travel or in its central position, D being the location of the crank pin.

Again referring to Fig. 90, draw line CA in such a position that the angle ABC will equal the angular advance of the eccentric, which we will assume in this case to be 30°.

This will bring the high point of the eccentric at B, while the crank pin, as before stated, is at D. Next, using line CA as the diameter, draw a circle about it, called the valve circle. Now, suppose the crank to be turning in the direction of the arrows. At position D the crank line is just about to cut into the valve circle, the valve being central. When the crank gets to position 1 the valve has moved the distance CE. When the crank is at 2 the valve has moved the distance CM, and when the crank arrives at 3 the valve has moved to the limit of its travel from its central position, and it now begins the return movement. The motion of the valve is comparatively slow at this point for the reason that the high point of the eccentric is now passing the center at 7. The distance the valve has moved backward while the crank has moved from 3 to 4 is the distance BF, while FC represents its distance from the central position, and GC the same when the crank is at 5. When the crank arrives at 6 and its line has left the valve circle, the valve is again central. Fig. 90 merely shows the movement of

the valve through one-half of its travel without giving any details regarding port openings, cut-off, etc.

In Fig. 91 the influence of outside lap is delineated. According to the dimensions of the valve under consideration the outside lap is one inch. The diagram is drawn

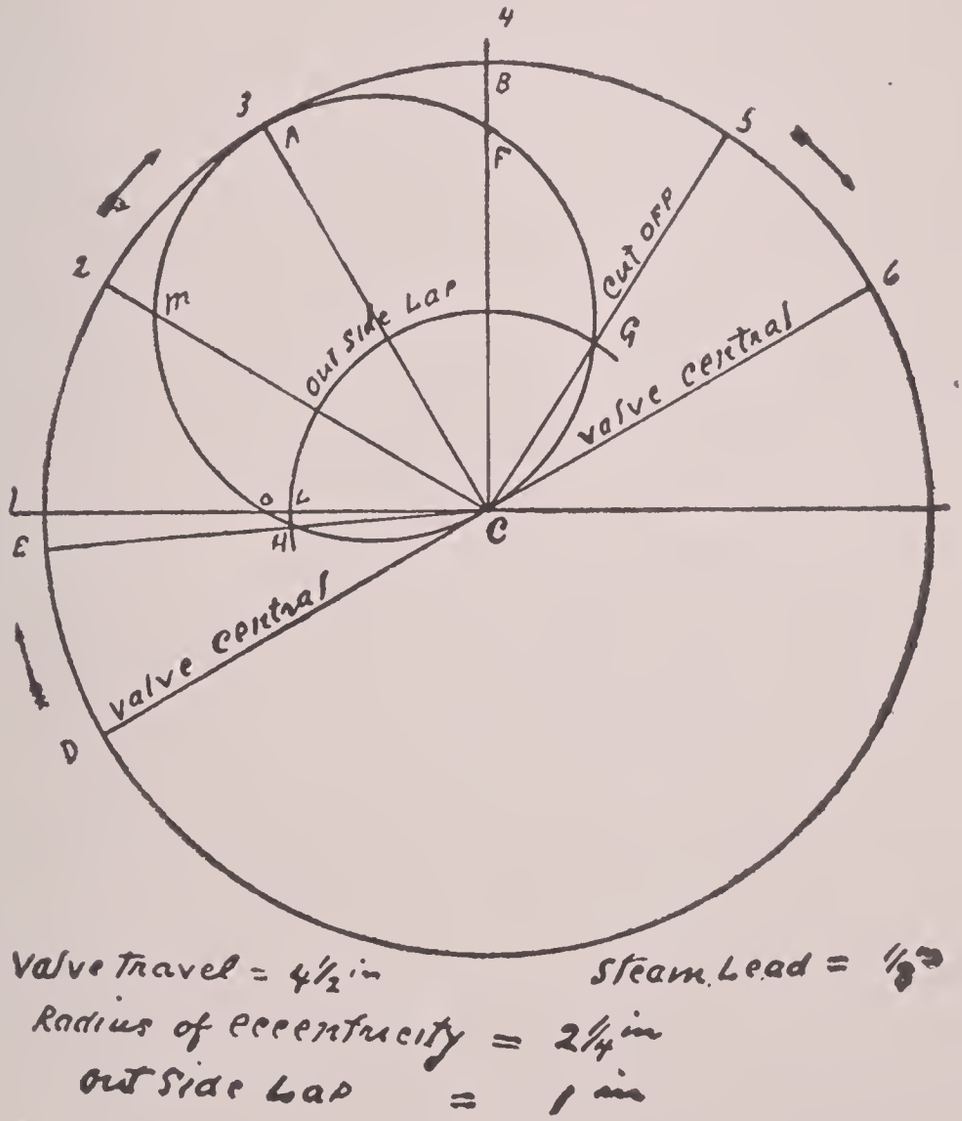


Fig. 91.

precisely as in Fig. 90, and in addition strike an arc representing the outside lap, using C as the center with a radius equal to the outside lap. As before, the crank is at D and the valve central. When the crank has moved to E and its line cuts the intersection of the outside lap

and valve circles, the valve has moved the distance CH, just equal to the outside lap, and the port begins to uncover at this point. Then by the time the crank gets to the center, 1, the port is open the distance LO, which is the lead, in this case 1/8 inch.

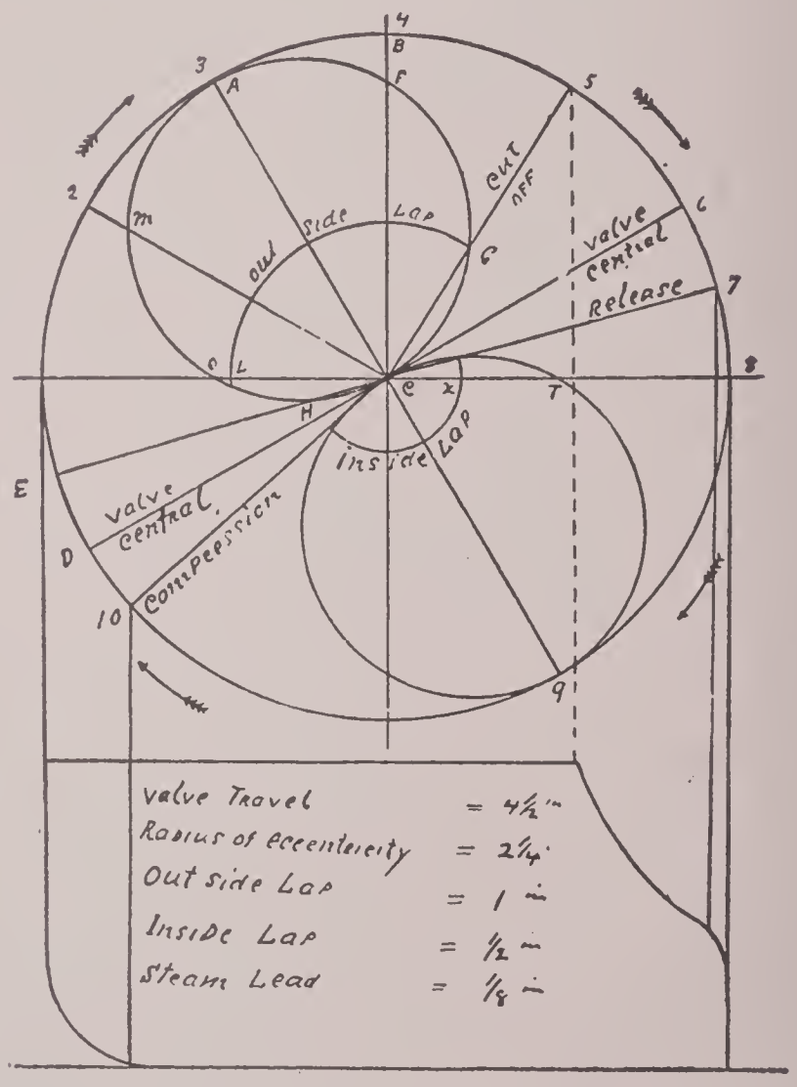


Fig. 92.

The position of the crank when cut-off takes place is ascertained by drawing a line, CG5, through the intersection of the outside lap and valve circles, where the valve is on its return movement. Thus far no account has been taken of release and compression, and in order

to determine the position of the crank when these events occur it will be necessary to draw the valve circle for the opposite movement of the valve, for, be it remembered, that the movement of the valve so far considered has been only one-half of its travel, that is, it has moved from its central position towards the head end of the cylinder, and back again. We have seen how it has performed the functions of admission, full port opening, and cut-off for the crank end of the cylinder, and now by reference to Fig. 92 it will be seen at what points of the stroke the remaining events, viz., release and compression, occur.

Draw a second valve circle, Fig. 92, diametrically opposite the first. Also draw an arc with the radius equal to the inside lap, which in this case is assumed to be one-half inch. When the crank gets to the position 7, its center line cuts the intersection of the inside lap and valve circles and release begins. When the crank arrives on the center 8, the valve has moved the distance CT from central position; but CX of this distance has been occupied by the inside lap, therefore the lead on the exhaust is represented by the distance XT. When the crank on its return stroke arrives at the position marked 10, its line again cuts the intersection of the inside lap and valve circles and compression takes place. By dropping perpendiculars from the positions of the crank at 1, 5, 7, 8 and 10, an indicator diagram may be drawn showing the performance of an engine with this style of valve.

Fig. 93 shows the effect of decreasing the angular advance, that is, setting the eccentric back towards the crank.

In this instance the eccentric is set back 10 degrees, thus making the angle of advance 20 degrees instead of

30 degrees as before. The full lines represent the new angle, while the dotted circles and lines indicate the valve, and its movements as drawn at first.

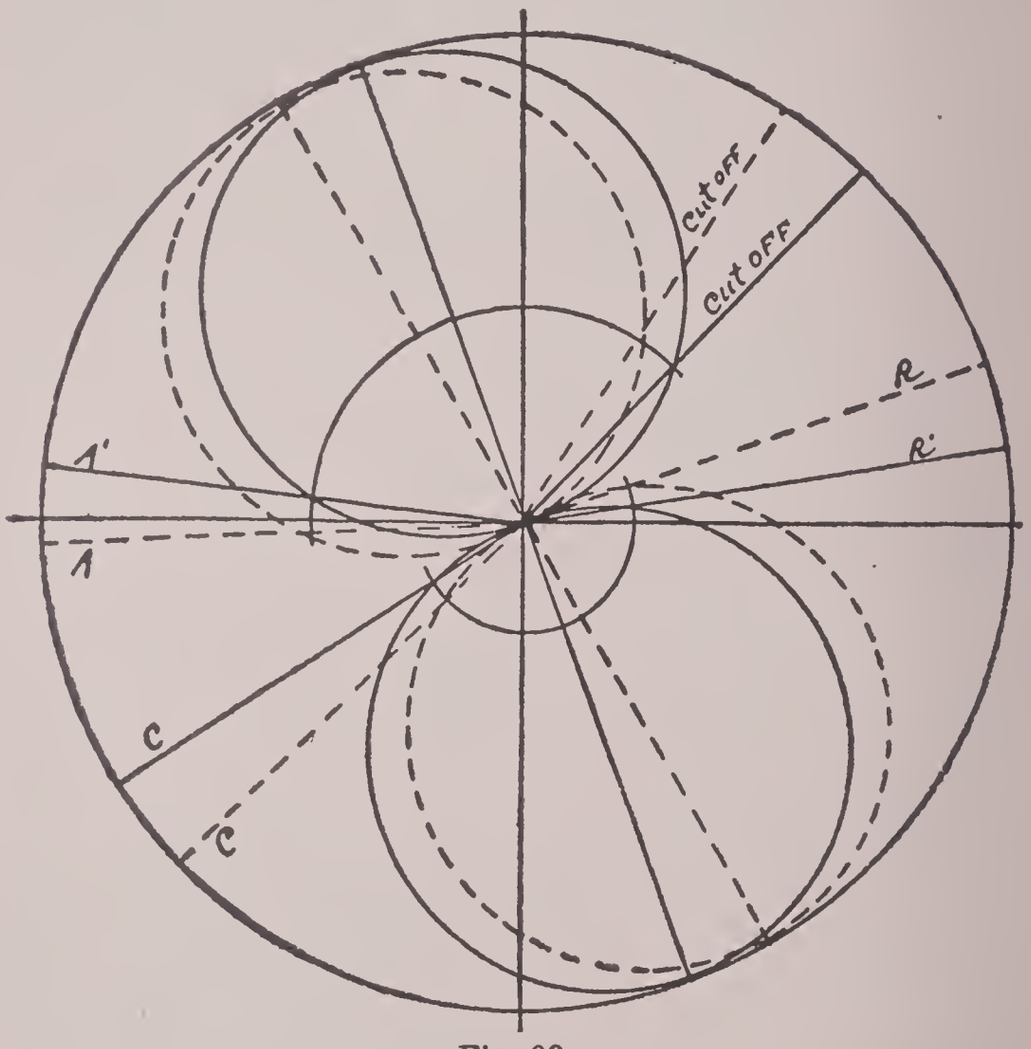


Fig. 93.

A shows the original point of admission, and A′ the position of the crank when admission takes place with the lesser angle of advance. Similarly R and R′ show the old and new points of release, and C and C′ the compression. The two different points of cut-off are also indicated.

The effect of decreasing the valve's travel is illustrated by Fig. 94, the full lines showing the decreased travel and

its influence, and the dotted lines showing the original. Admission and release occur later, while cut-off and compression take place earlier.

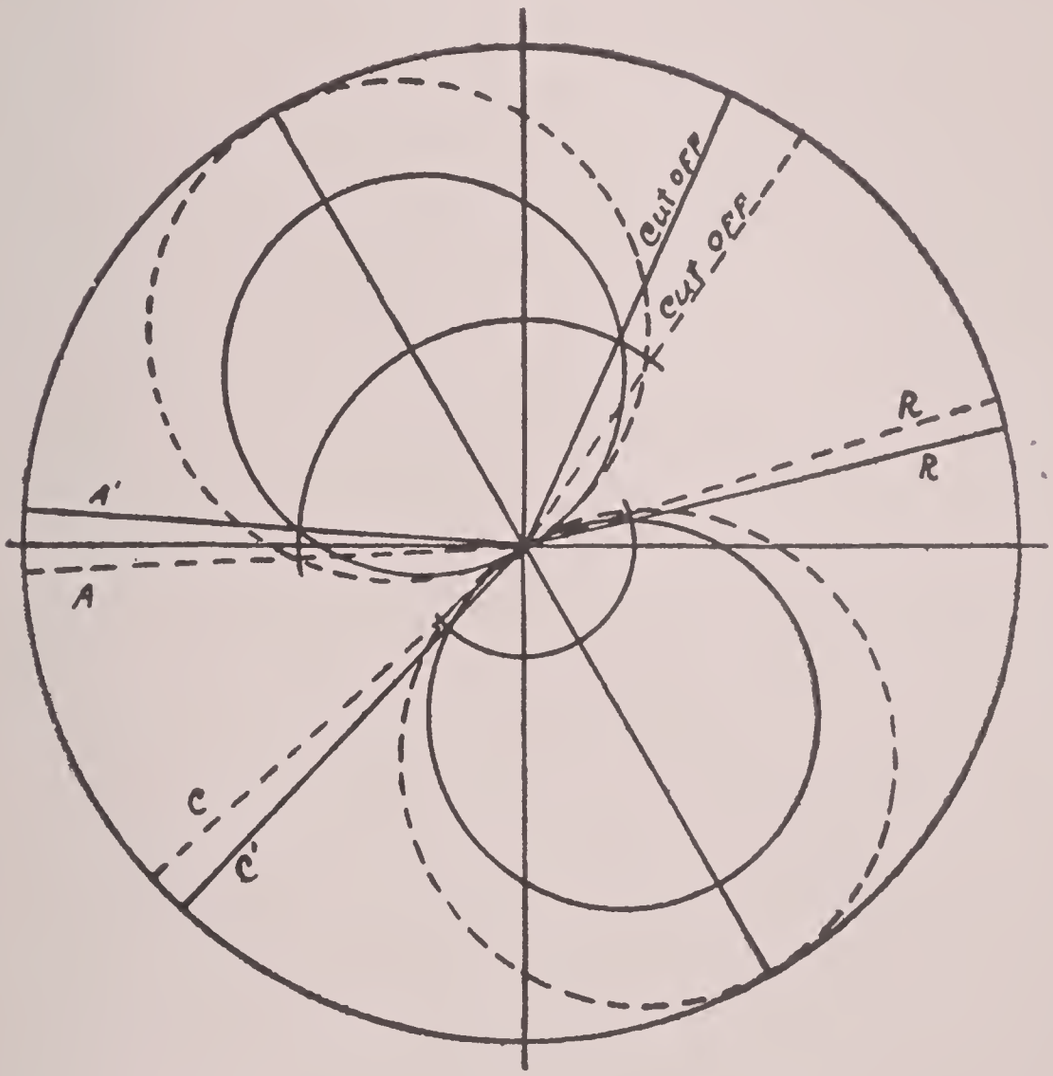


Fig. 94.

The travel of the valve as indicated in Fig. 94 has been decreased one inch, thus making it 3½ in place of 4½ inches as before. Fig. 95 shows the result of increasing the outside lap. The lap has been increased in this instance from 1 inch, as originally drawn, to 1¼ as indicated by the full lines, while the dotted lines show the lap as it was before being changed. The effect of this

change is to cause less lead, a later admission and an earlier cut-off, but compression and release are not affected, for the reason that these latter events are controlled by the inside lap, which has not been changed.

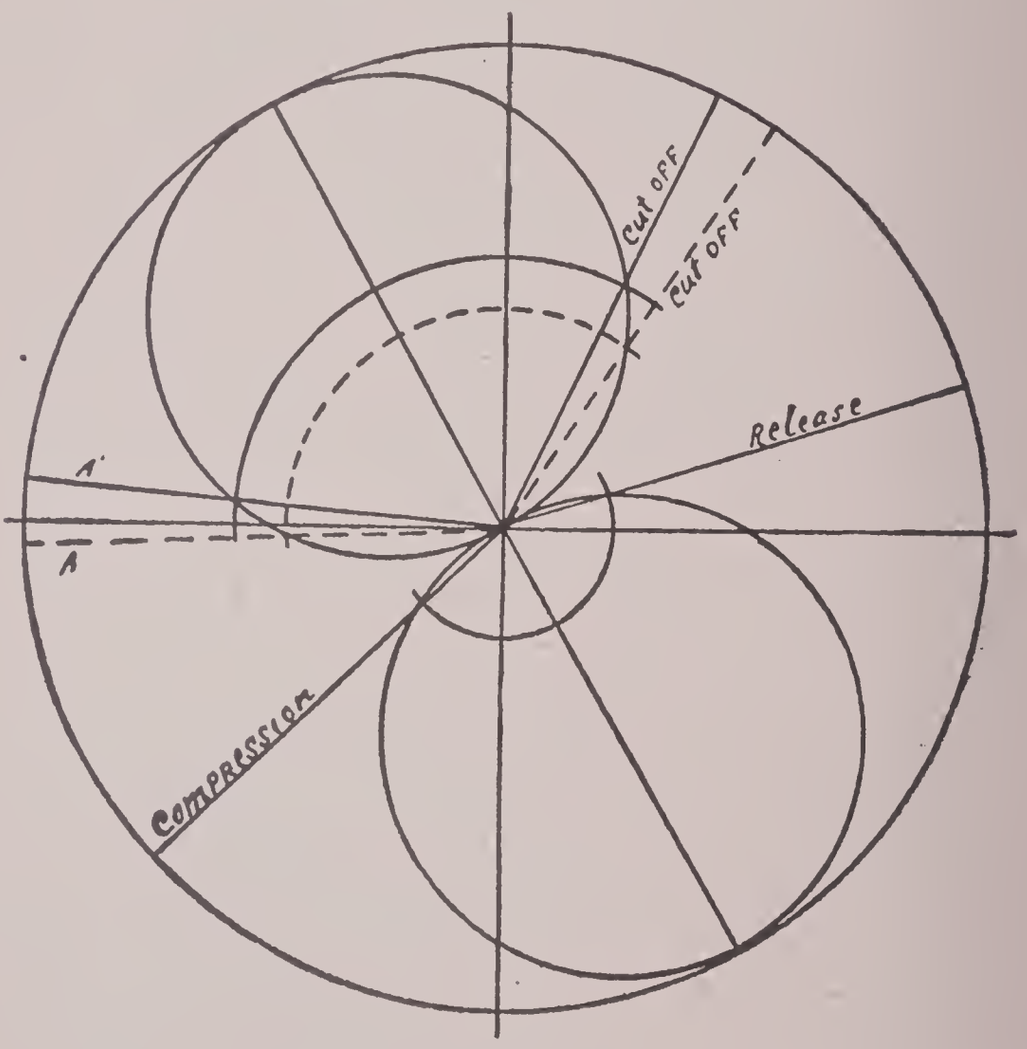


Fig. 95.

In Fig. 92 the valve is shown as cutting off the steam when the crank has completed 120 degrees or two-thirds of the half revolution, but the point of cut-off on the indicator diagram shows that the piston has traveled 7-9 of the stroke.

This discrepancy is due to the obliquity of the connecting rod, as it will be seen by looking at the valve diagram (Fig. 92), that the crank must travel farther to complete the stroke from this point than the piston does.

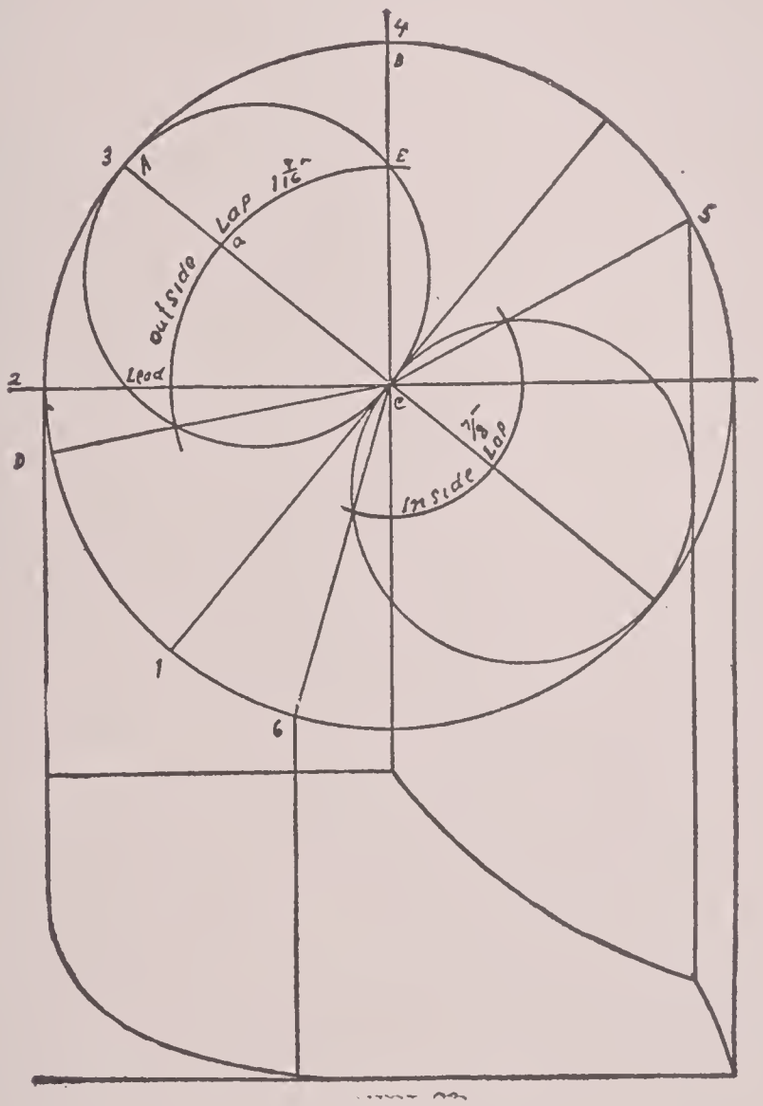


Fig. 96.

In order to cause the valve to cut off earlier, say at one-half stroke, it will be necessary to do one of two things, either to increase the outside lap, which would have a tendency to cause admission to occur too late, or the angle of advance may be increased sufficient to cause

cut-off to take place at half-stroke, but to do this alone would cause admission to occur too early. Therefore the proper thing to do is to increase both the angle of advance and the outside lap. Fig. 96 shows how this can be done without decreasing the travel of the valve. The angle of advance, ABC, is now 50°, where before it was 30°, as in Fig. 92.

The valve is central when the crank is at position 1; the high point of the eccentric being at point 4. The outside lap, which was before 1 inch, had 7-16 inch added to it, making it 1 7-16 inches. When the crank gets to D the port is just commencing to open, and with the crank on the center at 2, the lead is ¼ inch.

It will readily be seen at this point that by increasing the outside lap still more the lead can be diminished, and the point of cut-off made still earlier, but this would result in a still further reduction of the power of the engine, which has already been considerably reduced, as shown by the diminished area of the indicator diagram as compared with the one in Fig. 92. When the crank gets to position 3 the valve has reached the limit of its travel, and the port is open the distance Aa, which is as far as the outside lap will permit. With the crank at point 4 cut-off occurs. But with the increased angular advance and the inside lap remaining as it was before, viz., ½ inch, release would occur too early. Therefore it will be necessary to increase the inside lap sufficient to cause release and compression to take place at as near the proper points as possible. In this instance ⅜ inch has been added, making the inside lap ⅞ inch, and release takes place with the crank at position 5, while compression begins at 6.

These points, viz., release and compression, may also

be changed by simply adding to, or decreasing the inside lap. It should be noted that in the foregoing discussion of valve gear it is understood that the valve stem moves in the same direction as the eccentric rod does, that is the direction of motion is not reversed by a rocker arm interposed between the eccentric and the valve. In other

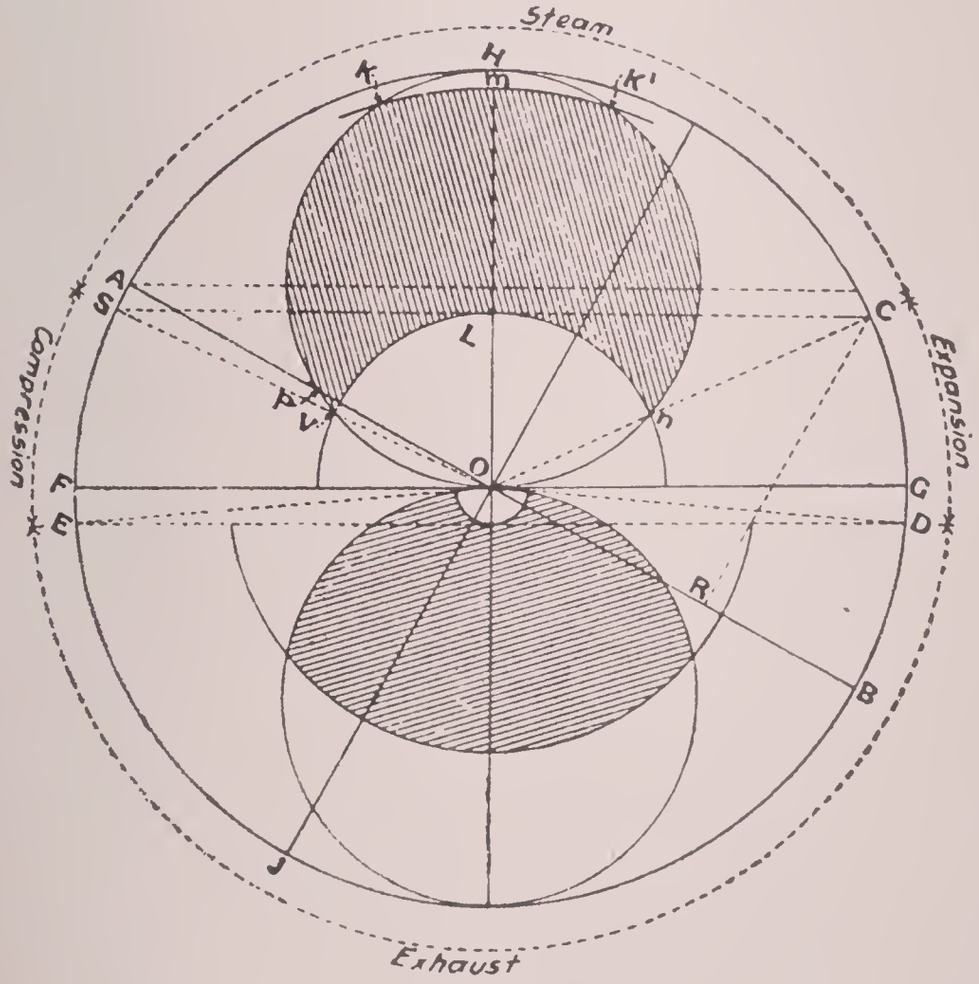


Fig. 97.

words, the motion is direct. In case the motion is reversed by a rocker arm (indirect motion) it will be necessary to set the eccentric behind the crank.

Fig. 97 is a combination of the Releaux and Zeuner diagrams used to represent the different valve events.

This cut, together with Fig. 98 and the explanation

following is presented by Mr. Carl J. Mellin, consulting engineer for the American Locomotive Co., and if closely followed will serve to still more clearly fix in the mind of the student an understanding of the action of the valve in the performance of its various functions.

Referring to Fig. 97, the distance AB represents the travel of the valve as well as the stroke of the engine, though in different scales, which makes no difference when the cut-off is always expressed in fractions or per cent of AB. The maximum cut-off is determined upon to be AR. Draw a perpendicular line RC from AB until it cuts the arc ACB. Next decide on a desired lead and with that as a radius, draw an arc with A as a center. Draw a line, from C tangent to the lead circle around A, when the lap of the valve is found to be equal to the perpendicular distance from the line CS to the center O of the diagram. The crank will then be in position OS when the valve commences to open, or the angle AOS in advance of the dead center, and on OC at cut-off. Continuing, we find the valve in its middle position when the crank is on OG which is drawn parallel to SC through the center O. Extend this line to F, and with the exhaust lap as a radius draw the exhaust lap circle on the opposite side of the line GF and draw DE tangent to this circle, when OD is the position of the crank at the release point. From this point the exhaust remains open until the crank reaches the position OE, when it closes and compression takes place until it again reaches OS for admission and one revolution is completed.

By placing the Zeuner diagram upon this, draw HJ perpendicular to FG, and with the radius OH of the eccentric circle as a diameter, draw the admission valve circle OVHnO, and the lap circle with the steam lap as a

radius and find the intersection occurs at V, both with the circles and the previously laid down admission line OS and the cut-off point at the intersections at n. On the line OH set off the width of the steam port from L toward H equal to Lm and with Om as radius draw the arc KmK^1. The shaded figure enclosed by the letters

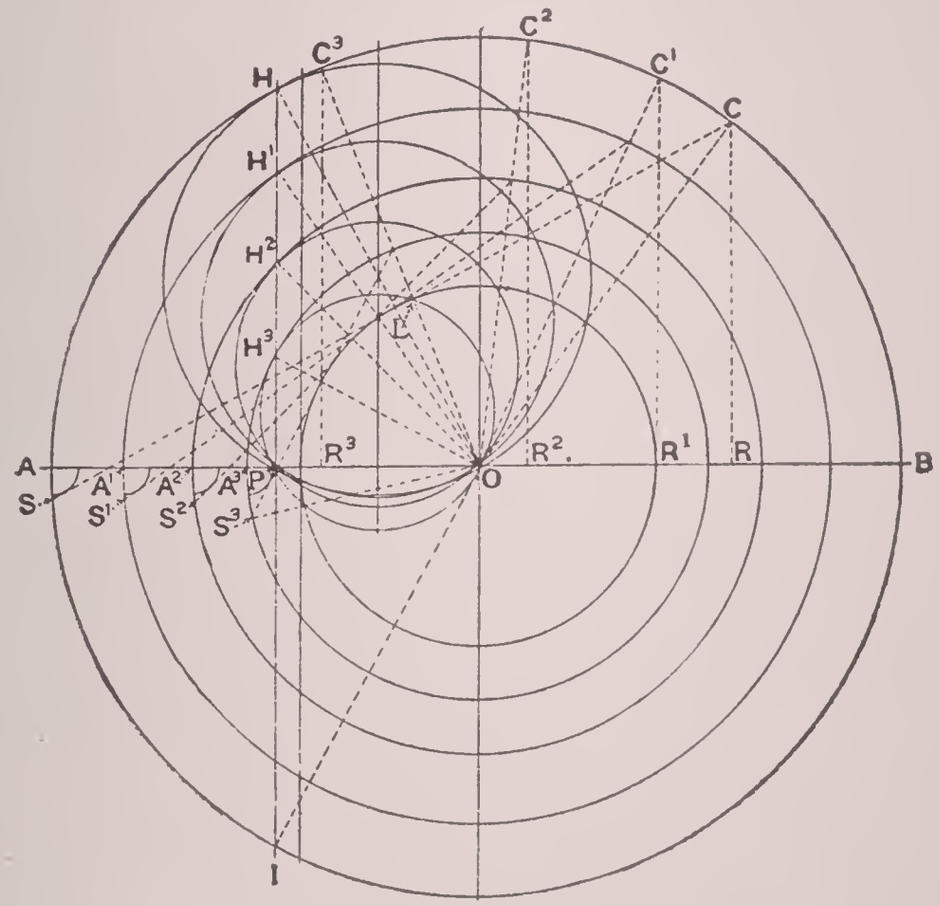


Fig. 98.

VKK^1nL represents the steam port opening during the admission period and the width of the port opening at any desired position of the crank is found by measuring the distance radially from O between the lap and valve circles on the port line, as the case may be, on the desired crank position.

The exhaust openings are determined in the same manner and are shown on opposite side of FG, where the crank passes through the arc DJE during the exhaust period with a positive exhaust lap of the size EF. When the exhaust edge of the valve is line and line this arc becomes GJF or 180 degrees, and when a negative lap (clearance) occurs, the duration of the exhaust period exceeds the half revolution of the crank. The various events are indicated around the eccentric circle on the figure as they take place during a complete turn of the crank.

In Fig. 98 the eccentric and admission valve circles are shown at different cut-offs where each set of lines and circles is governed by the same explanation as those of Fig. 97, where the admission points S, S^1, S^2, S^3 correspond to the closing positions C, C^1, C^2, C^3, cut-off points R, R^1, R^2, R^3, etc. On OH we have the full travel valve circle and OL the lap or radius of the lap circle, the latter being the same for all cut-offs as well as the lead, the radii H^1, H^2, H^3, etc., of the eccentric circles or diameters of the corresponding valve circles terminate on a line HI drawn perpendicular to AB and at a distance from O equal to that of lap and lead.

When the reverse lever is in its center position the diameter of the valve circle falls on the line AB and is equal to lap and lead. Continuing in back position we have the same method repeated and OI would be the full travel valve circle diameter, or the same as the eccentric radius for the valve travel. Any desired cut-off position may be laid out in same manner as that in Fig. 94, which shows all the valve events for a complete revolution of the axle.

The movements are in actual practice not so regular

as the circles indicate, as it is impracticable to obtain the various loci in their theoretical positions; besides we have the angularities both of the main rods and the eccentric rods to contend with, and whereby irregularities are entering in the problem that must be compensated

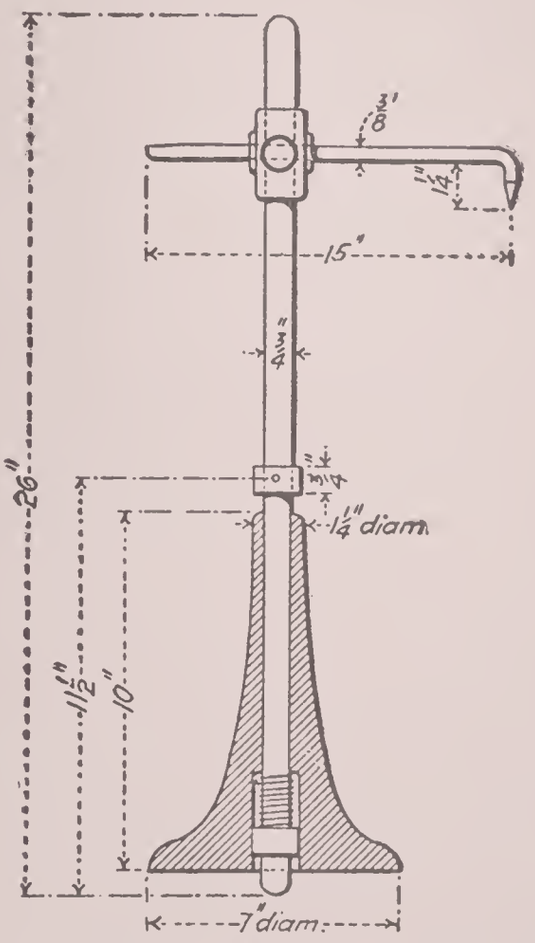


FIGURE 99. A CONVENIENT GAUGE FOR SETTING THE ECCENTRIC CRANK.

for, as referred to in the general description. It is not to be considered that a uniform circular motion is the best, but an approximation to it works with less shocks or jerks, and is therefore more desirable for so high speed an engine as a locomotive. A few advantages can be taken, however, in selecting the suspensions and various connections, so that better results can be obtained

than from a true circular motion, which are principally affected by three union points and are, first, the connecting point of eccentric rod and link; second, the locus of the lifting link suspension point; and third, the relative height of the crosshead connection point of the union bar to the corresponding point of the combination lever.

It is not necessary to lay out the valve diagrams except where a given cut-off per cent. is wanted. This is the most convenient way to find the required lap.

Reason for Increase of Lead as Reverse Lever is Locked Back with Stephenson Valve Gear.

The explanation of this problem, although simple enough, is generally presented to the student by technical writers in such a form as to be incomprehensible to the man with limited education, therefore the author considers it his duty to explain in as clear a manner as possible the causes of this very annoying, but inevitable occurrence.

Lead is the amount of opening that the valve gives to the steam port when the engine is on the dead center, that is with the piston at the extreme end of the stroke, and this lead is obtained by advancing the eccentric the required number of degrees ahead of the crank in the direction it is to run.

Referring to Fig. 100, let a, represent the locus of the eccentric and eccentric strap with the reverse lever in the position of full valve travel. Now as the lever is hooked back to shorten valve travel the eccentric rod is raised, and the eccentric strap is moved back on the eccentric, which would increase the lead to the same amount as moving the eccentric ahead on the driving axle would. Mark a, would now be broken, as at b.

The amount that the lead is increased depends to some extent upon the length of the eccentric rods, as with long rods the lead will not be affected so much, as with short rods.

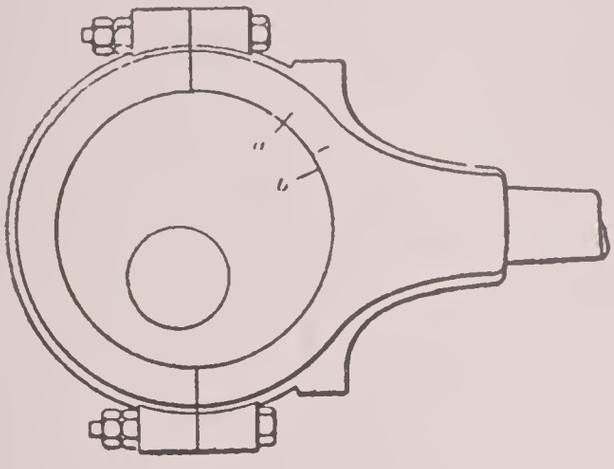


Fig. 100.

VALVE SETTING.

The correct setting of the valves of a locomotive means that the adjustment of the positions of the eccentrics on the driving axle and the lengths of the eccentric blades, valve rods and valve stems is such that each valve will give the required distribution of steam to the piston that it is to serve. This has already been explained under the heading, Function of the Slide Valve. As the great majority of locomotives are equipped with indirect link valve gear, attention will now be directed to the setting of valves operated with this type of valve gear. One of the first things to be done is to see that the driving wedges are properly adjusted, also that the main rod keys at both ends are correctly tightened. It is also well to see that the eccentric rods are connected in the right way, which means the go-ahead eccentric rod to the top end of the

link, and the back-up eccentric rod to the bottom end of the link.

Don't forget that with the indirect link motion the eccentric that controls the valve always follows the crank pin. That is, when the pin is on the forward center, for instance, the body of the go-ahead eccentric will be above the axle, and that of the backing eccentric will be below, and both eccentrics will be advanced towards the pin sufficient to overcome the lead and lap of the valve. This is termed angular advance of the eccentric. The eccentrics should be placed as near as possible in these positions and the set screws slightly tightened. Of course the positions of the eccentrics can only be guessed at on the start. Their correct positions on the driving shaft can only be arrived at after the dead centers have been located. The reverse lever should also be tested to see that the latch will enter each extreme notch.

The next most important proceeding is to get the port marks properly located on the valve rod. This, of course, must be done while the steam chest cover is off. The valve stem key should be examined to see that it is securely tightened. Next examine the back end of the valve rod and see that it will connect with the rocker arm without cramping or twisting the valve stem, which would be liable to throw the ends of the yoke up or down, thus cramping the valve. As the steam chest cover is off, the chest itself should be firmly clamped to the cylinder by screwing some of the nuts down upon washers or bushings, being careful not to mar the copper joint on top of the chest. The valve stem gland should be in place, and the valve rod connected up in order to keep the stem at its proper height, as any variation in the height of the stem will cause an error in the use of the tram. If there is any

lost motion between the valve and the valve yoke (and there should be a little), it should, while getting the port marks, be taken up by the use of liners between the back of the valve and the yoke, as shown at A in Fig. 101. Next move the valve back just far enough to permit a piece of thin tin to be inserted between the edge of the

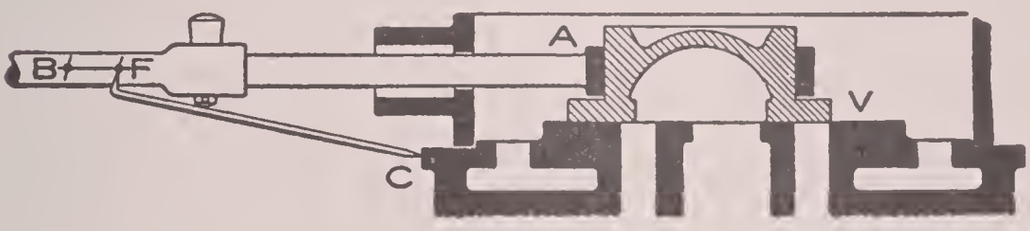


Fig. 101.

forward port and the forward edge of the valve at point V. The valve is now in the correct position for the forward port mark to be placed upon the valve rod; so with a prick punch make a small center at C on the cylinder,

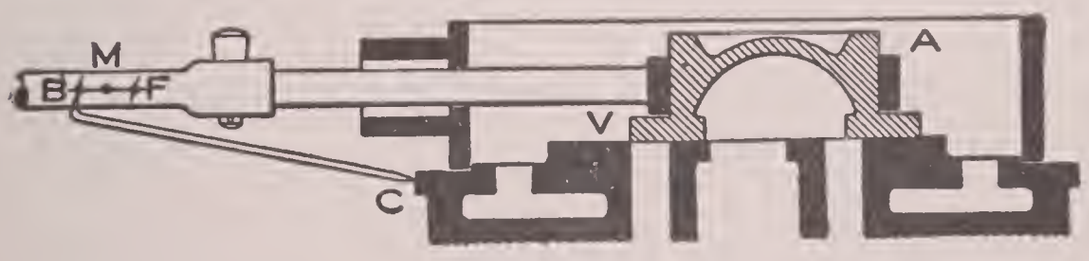


Fig. 102.

and from this point with the valve tram, as shown in the cut (Fig. 101), scribe the line F on the valve rod. Next remove the liners and place them at the front of the valve at A, Fig. 102, and move the valve ahead far enough to allow of the insertion of the piece of tin between the edge of the back port and the back edge of the valve, as shown at V, Fig. 102. Now take the tram and from point C scribe the line B on the valve rod. Next, using a box square, scribe a parallel line on the valve rod, and

at the two points where lines F and B intersect the parallel lines, make two small centers. Center F is the forward port mark and center B is the back port mark.

The mid-travel or central position of the valve on its seat should also be marked, which is done in the following manner: With a pair of dividers find the exact center between points F and B and at this point make another small center M, Fig. 102. This point will represent the central position of the valve. The points F and B indicate the points of admission and cut-off, and the distance from F to M or from B to M equals the lap of the valve. If the valve has neither inside lap nor inside clearance the point M will represent the points of both release and compression. If the valve has inside lap or inside clearance it will be necessary to locate two additional points on the valve rod. These points may be found in the following manner: Set a small pair of dividers to distance equal to the inside lap or inside clearance, whichever the valve has, and from the center M describe a small circle, and the two points where the parallel line on the valve rod bisects this circle will indicate the points of release if it is inside clearance or of compression if it is inside lap. These two points are very seldom used in practice, but if, owing to the construction of the valve, it should become necessary to use them it should always be remembered what they represent, whether inside lap or inside clearance.

The next important move in valve setting is to find the four dead centers. It is very important that the dead centers be accurately located. Although the crosshead moves very little while the crankpin is near the dead center, yet the valve is moving at nearly its greatest speed, being at about half travel, and a very slight error

in locating the dead centers will seriously affect the accuracy of the whole work.

The term dead center is commonly taken to mean that the driving wheels are in such a position that the centers of the driving axles and the centers of the crankpins are in a horizontal line, but this is not always the dead center.

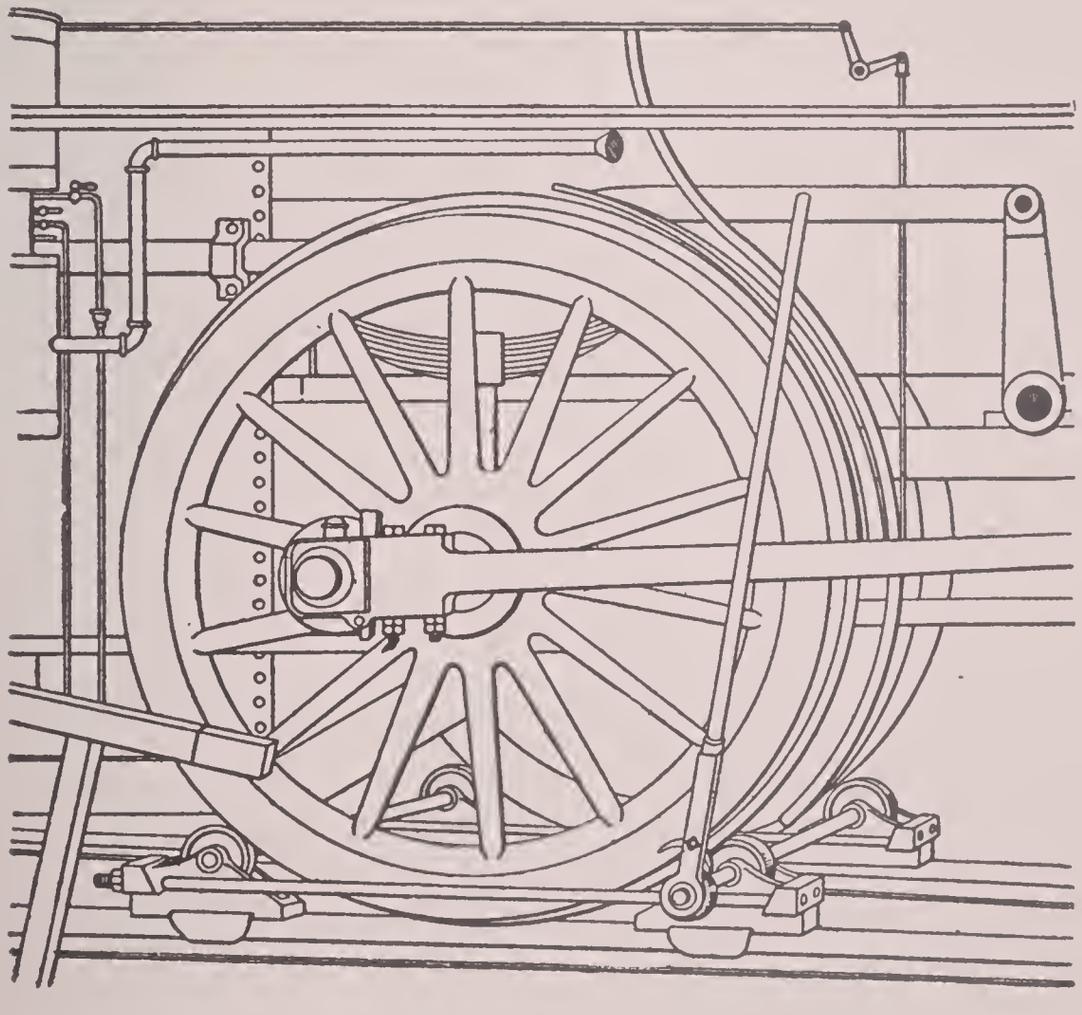


Fig. 103.

Theoretically the term implies that the center of the crosshead pin, the center of the crank pin, and the center of the driving axle be exactly in line, regardless of whether that line be horizontal or inclined. Therefore the crankpin must pass two dead centers in each revolution, viz.,

the forward dead center and the back dead center. Consequently there are in locomotive valve setting four dead centers to be located and marked: first, the right forward dead center; second, the right back center; third, the left forward center, and the fourth, the left back center. It makes no particular difference which center is found first, but for convenience the right forward center may be taken. Of course finding the dead centers of a locomotive implies that the driving wheels are to be revolved more or less. This means that the engine must be pinched ahead or back as required, which involves considerable labor on the part of helpers, as many an engineer who has served an apprenticeship in the shop can testify. Many well conducted shops are equipped with roller devices of various designs which are placed under the drivers to be revolved. Such a machine is illustrated in Fig. 103. It may be operated by one man by means of the lever shown. In some up-to-date shops the rollers for moving the drivers are operated by small air engines.

And now to find the right forward dead center. Turn the driving wheels forward until the cross-head is within an inch of the forward end of the stroke, as indicated in Fig. 104. Then, having first examined the wheel cover to see that it is securely fastened, make a center at any convenient point on it, as at C; also make a center at point F on the forward guide block. Now, using a short tram called a cross-head tram, describe from point F an arc G on the cross-head; also with one point of a longer tram, called a wheel tram, set in center C describe the arc A on the tire of the wheel. Next turn the wheel ahead as indicated by the arrow until the cross head has passed the limit of its forward travel and has receded

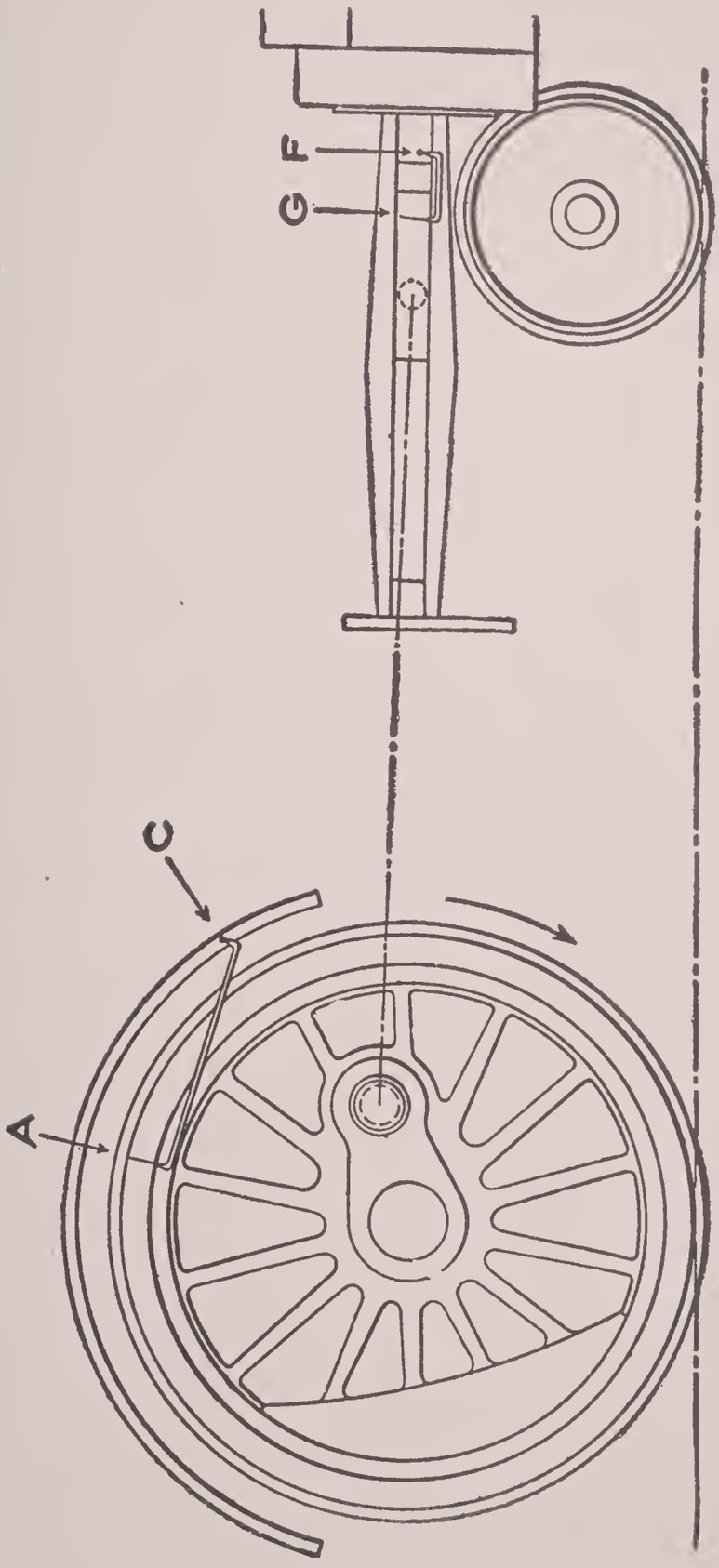


Fig. 104.

on its stroke far enough to bring the arc G a short distance back of the point of the tram, one point of which is set in center F. Now reverse the motion and pinch the wheels slowly backward until the arc G comes directly under the point of the tram. Then stop, and with the wheel tram set in center C scribe an arc B on the tire, as indicated in Fig. 105. Now with a pair of hermaphrodites describe the arc D E on the tire, and at the points where the lines A and B intersect arc D E make small centers, and with a pair of dividers find the exact center between these two points. This center is indicated in the cut (Fig. 105) by the letter H. This point is the dead center, and a small circle should be drawn around it to distinguish it from the other centers.

Perhaps the query might arise in the mind of the student, why is it that in turning the wheels ahead until the pin had passed the center they were turned far enough to bring the cross-head back of the position it was in when arc G was scribed? The answer is, that when arc G was scribed the pin was pushing the cross head forward and all the lost motion between pins and brasses was taken up in that direction. If, after the pin had passed the center and the cross-head was traveling back, it had been stopped at arc G, the lost motion would have been taken up in the other direction, for the reason that the pin was now pulling instead of pushing the cross-head. The result of this would have been an error in the location of the arc B and also of the point H. But by pulling the cross-head back past arc G and then reversing the motion and allowing the pin to push the cross-head until the dead point H was located, the lost motion was taken up in the same direction as when arc G was first drawn.

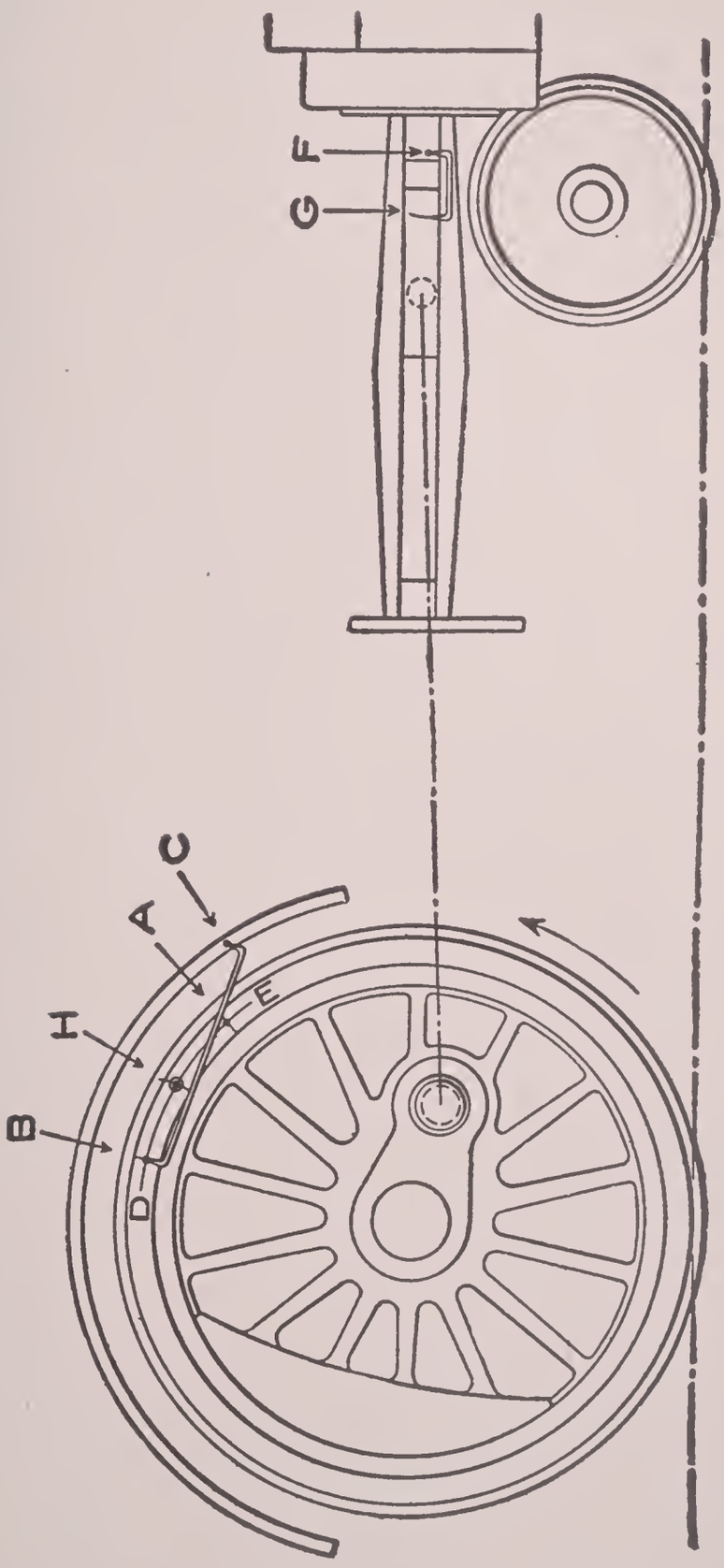


Fig. 105.

Having now found the dead center at point H, the next move is to throw the reverse lever into the extreme back notch, so as to take up all the lost motion in the valve gear while backing up. Now start to pinch the engine back, and with one point of the wheel tram in center C, watch the center H and when it comes exactly under the other point of the tram, stop. The engine is now on the right forward dead center, and a vertical line should be scribed on the guides exactly in line with the front end of the crosshead. This line indicates the extreme forward travel of the crosshead, and it is important that it should be placed there.

While the engine is in the position it now is, that is, on the right forward dead center, and the valve gear in the backward motion with all the lost motion taken up in that direction, take the valve tram and from point C, Fig. 101, scribe an arc on the valve rod, starting slightly above the parallel line and extending considerably below it. The distance of this arc, measured on the parallel line from center F, indicates the position of the valve, as regards lap or lead for backward motion. The reason this arc is drawn below the line is that the back-up eccentric is moving the valve, and by having the arc below the parallel line it is easily distinguished from the other arc soon to be scribed for the forward motion. Now pinch the wheels back until the crankpin is about 6 in. above the dead center. Then put the reverse lever in full forward motion and pinch ahead until the pin is again on the forward dead center, and with the valve tram again set in point C scribe another arc on the valve rod, this time extending above the parallel line. The distance this arc is ahead or back of the point F indicates the amount of lap or lead the valve has in the forward

motion, when the crankpin is at right forward dead center.

Before making any adjustments, go round to the left side of the engine and find the left forward dead center; also mark the left valve rod for both forward and back motion in the same manner as the right valve rod was marked. Having completed the location of the forward dead centers for both sides, the next move is to start on

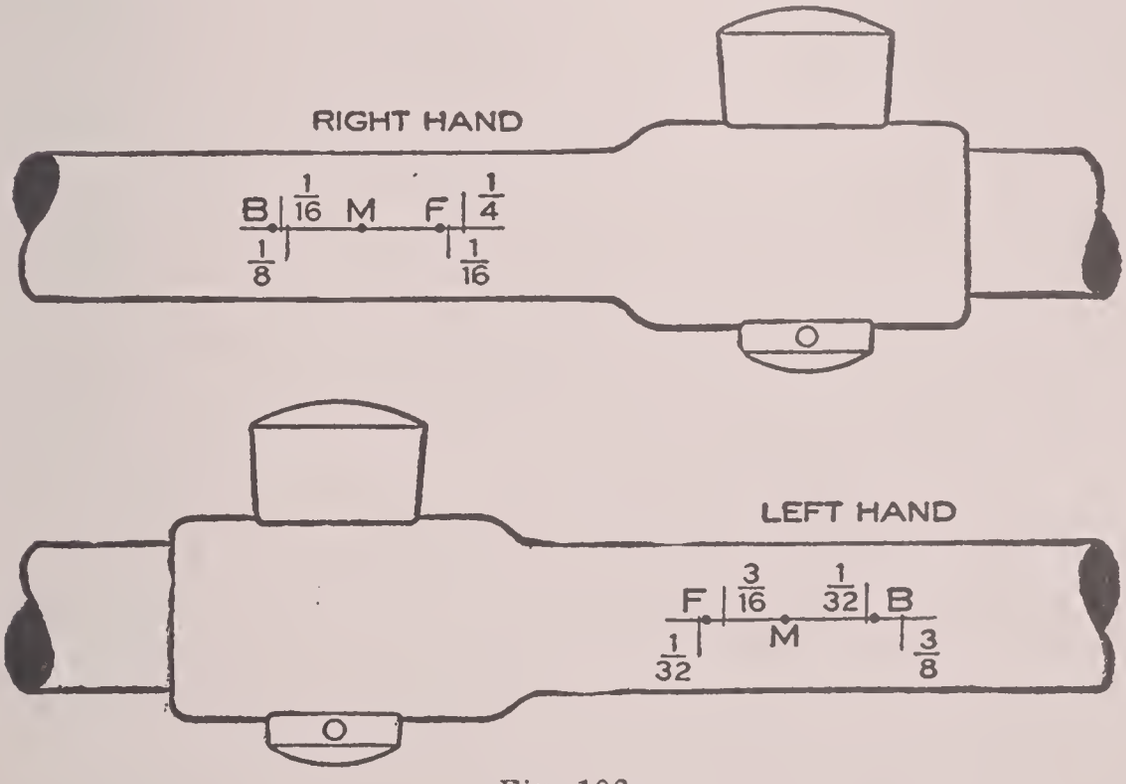


Fig. 106.

the right-hand side again, and pinch the engine towards the right back dead center, which is to be found in the same manner as the forward one was. Next find the left back dead center, marking the valve stems and cross-heads in both instances exactly as before.

The valve rods will now show a marking similar to that illustrated by Fig. 106, with the exception that the figures may not coincide, as the figures shown in Fig. 106 are merely assumed for purposes of explanation.

As before stated, the arcs which have been scribed across the parallel lines indicate by their position relative to the port marks F and B whether the valve has lap or lead at either dead center. If an arc comes between the port marks it indicates lap, if outside it indicates lead. Referring to Fig. 106, the two forward motion arcs on the right side valve rod, which are distinguished from the back motion marks by being above the parallel line, show that the valve has ¼-in. lead at the forward port mark and 1-16th in. lap at the back port mark.

When the valve tram reaches from center C to either center B or center F, it indicates that the valve is at the point of cut-off, and since the valve is to travel equal distances each way from these points, it follows that by measuring the distance from B or F to the arcs, it may be determined how much and whether to lengthen or shorten the eccentric blades. First take the right forward motion. The distance from F to the mark above the line is ¼ in., and from B to the mark for back motion is 1-16th in., therefore the length of the right forward motion eccentric blade must be changed so as to equalize these distances, and the point to be determined is, shall it be lengthened or shortened? This can be done in the following manner: Take a small pair of dividers and find the exact center between the two tram marks above the parallel line. If this center is ahead of center C the eccentric blade must be shortened, if back of it the blade must be lengthened. If the engine has a direct valve motion, this rule is to be reversed and the adjustments made accordingly.

The next point to be determined is, how much shall the blade be lengthened or shortened? A good rule to follow in this instance is this: When the arcs on the

valve rod are both back or both ahead of the port marks F and B, the length of the eccentric blade should be altered an amount equal to one-half the sum of the distances between the port marks and the arcs, or if one arc is back and the other is ahead of their respective port marks, the length of the blade should be changed an amount equal to one-half the difference of the distances between the port marks and the arcs. In this particular case the valve has traveled too far back, as shown by the ¼-in. lead on the forward port mark and the 1-16th-in. lap on the back port mark. Therefore the blade must be shortened one-half the sum of these distances, or $\frac{\text{¼} + \text{1-16th}}{2}$ = 5-32nd in. This will square the valve for right forward motion, or in other words equalize its travel in either direction from mid position, as may be proved by the following simple calculation. The valve had ¼-in. lead at the forward port mark, the eccentric blade is shortened 5-32nd in., thus bringing the point of the tram that much nearer to F. Then ¼ — 5-32nd = 3-32nd in., which is now the lead at the forward end. At the back end, instead of lead, the valve had 1-16th in. lap.

After the blade is shortened 5-32nd in. it will be found that the valve has been moved that distance ahead from its former position. Then by deducting the 1-16th in. lap from 5-32nd in., it will be found that the valve has 3-32nd in. lead at the back end also. It may be assumed that the valves are to have 1-32nd in. lead when in full gear, and as the valve under consideration now has 3-32nd in. at both ends, it will be necessary to reduce it by turning the eccentric back upon the shaft. However, no changes should be made until all the tram

marks on both sides of the engine have been examined and a memorandum made of the changes required, as, for instance, R. F. Ecc., shorten blade 5-32nd in., 1-16th in. lead off.

The tram marks for the right backward motion should be examined next. These marks are below the parallel line, and measurements show that the valve has 1-16 in. lead at the forward end and ⅛ in. lap at the back; therefore the blade of the right back up eccentric must be shortened thus, $\frac{⅛ + \text{1-16th}}{2}$ = 3-32nd in. This will square the valve for right backward motion, but it will still have 1-32nd in. lap at both ends, when 1-32nd in. lead is required; therefore the eccentric must be turned ahead. These changes should be noted down as follows: R. B. Ecc., shorten blade 3-32nd in., 1-16th in. lead on.

If the upper and lower rocker arms are of the same length the figures for changing the length of the eccentric blades will be all right, but if, as is often the case, the lower arm is shorter than the upper one, the length of the blades will not need to be changed quite as much as is indicated by the marks on the valve rod. But it will be assumed in this instance that the arms are of equal length, and the lengths of the eccentric blades for the right-hand side may be adjusted according to the above figures.

Next go to the left-hand side. By reference to Fig. 106 it will be seen that the valve has 3-16-in. lap on the left forward motion in front and 1-32-in. lap behind. In this instance the valve has not traveled far enough back, therefore the blade must be lengthened one-half the difference between these distances or $\frac{\text{3-16th} - \text{1-32nd}}{2}$ = 5-64th in. This will equalize the lap at both ends,

making it now 7-64th in., and in order to obtain the 1-32nd-in. lead desired it will be necessary to move the eccentric ahead on the shaft an amount sufficient to overcome the 7-64th-in. lap plus 1-32nd-in. lead, a total of 9-64th in. This is to be noted down as follows: L. F. Ecc., lengthen blade 5-64th in., 9-64th-in. lead on.

Examination of the two left back motion marks shows that the valve has 3/8-in. lead at the back and 1-32nd-in. lead in front. Therefore the back up eccentric blade should be lengthened $\frac{3/8 + 1\text{-}32\text{nd}}{2} = 13\text{-}64\text{th in.}$ This will give the valve 13-64th-in. lead at both ends, but as 1-32nd-in. lead is all that is required, it will be necessary to turn the eccentric back on the shaft far enough to overcome 11-64th in. of this surplus lead. Note this down also as follows: L. B. Ecc., lengthen blade 13-64th in., 11-64th-in. lead off.

The lengths of all the eccentric blades should now be adjusted according to the figures obtained, after which it will be in order to set the eccentrics. It is generally best to set the forward motion eccentric first, because it is easier to get at than the back motion one is; then if the backward motion eccentric needs to be changed enough to affect the lead in forward motion, the forward motion eccentric can easily be reset, and it will need to be moved so little that the backward motion will not be affected enough to require any further attention.

As before stated, it is desired to give the valve 1-32nd-in. lead in full gear in both forward and backward motion, and before setting the eccentrics it will be necessary to have lead marks on the valve rods for a guide. To get these marks, set a pair of dividers to the distance between the centers B and F, Fig. 107, plus the lead,

in this instance 1½ in. + 1-32nd = 1 17-32nd in. Then with one point of the dividers in center F scribe an arc E across the parallel line, back of center B, also from center B scribe an arc D in front of F. These points, E and D, will serve as guides in setting the eccentrics for lead. The next move is to place the engine on the dead center; either one will do, but for convenience it may be

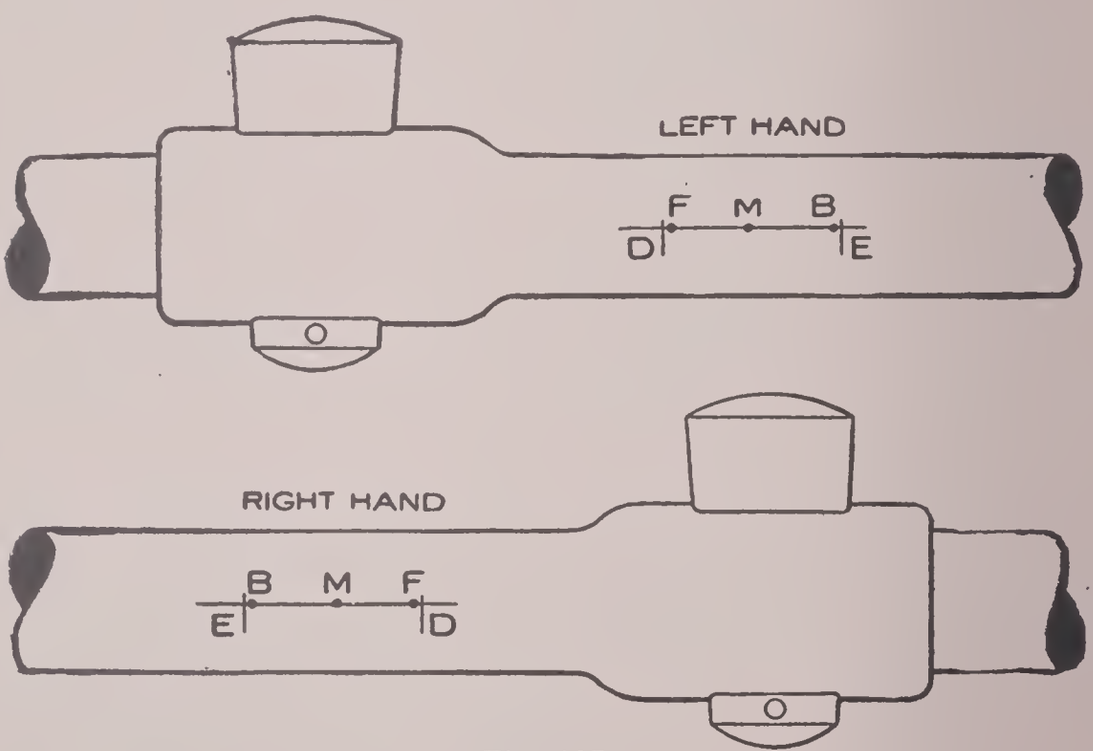


Fig. 107.

assumed that it is the right forward dead center. When adjusting the lengths of the eccentric blades it was found that with the engine and reverse lever in this position the valve had 3-32nd-in. lead. This must be reduced to 1-32nd in., and it might be done by simply turning the right forward motion eccentric back upon the driving shaft, but that would take up the lost motion in the opposite direction to what it is when the engine is running ahead, and this would cause an error in the working of the valve. The proper method is to turn the eccentric

backward far enough to take off all the lead, and then turn it slowly ahead until the valve tram will reach from center C, Fig. 101, to the lead mark D, Fig. 107. Next throw the lever into full back gear and proceed to set the right backward motion eccentric.

After getting the right backward motion eccentric blade adjusted to the correct length it was found that the valve had 1-32nd-in. lap at both ends; therefore this eccentric should be slowly turned ahead on the shaft until the tram will reach from center C to lead mark E, Fig. 107. This will square the right-hand valve and give it the desired lead, and the next move is to go around to the left side, throw the reverse lever into full gear ahead again and pinch the engine onto left forward center.

After adjusting the left forward eccentric blade to the correct length it was found that the valve had 7-64th-in. lap. The eccentric must therefore be turned ahead until the tram will reach from center C on the left cylinder to lead mark D on the left valve rod. Next proceed to set the left backward motion eccentric, and in doing this the wheels should be pinched ahead about 6 in., then place the reverse lever in full back gear and pinch the engine back onto the center. This is done to take up all the lost motion in the direction in which the engine is to run —a very important matter that should never be lost sight of in working from the dead center for either forward or backward motion. After getting the left backward motion eccentric blade the right length the valve had 13-64th-in. lead at both ends of the stroke. The eccentric should therefore be turned back sufficiently far to take off all the lead. Then with all the lost motion taken up, turn the eccentric slowly ahead until the tram point will drop into lead mark E, Fig. 107.

The engine is now square, and the valves have tne correct amount of lead all around. The eccentrics should be securely fastened in their proper location either by set screws or keys, and it will be next in order to ascertain the points of cut-off, so that they may be equalized as near as possible, for be it remembered that no matter how accurate the valves may have been set, as regards lead, travel, etc., they very seldom cut off the steam at the same distance from the commencement of the stroke at each end of the cylinder, and one cylinder may be getting more steam than the other. This is due to the fact that the link motion is not a perfect valve gear, various errors being introduced by the angularity of the main rod, and eccentric rods, and the off-set of the link pin holes from the link arc, but these errors can be almost entirely eliminated by making certain changes, among which may be mentioned the off setting of the link saddle stud, although with case-hardened links and the saddle rigidly bolted to the link this method is not always practicable.

Another very common method is to equalize the forward motion by changing the length of the backward motion eccentric blades, thus sacrificing equality of lead and cut-off in the back gear, but as a locomotive does the greater portion of its work in the forward gear, except it be a switch engine, this plan is permissible.

Another method employed to some extent is to sacrifice equality of lead in both forward and back gear for equality of cut off. But before either plan can be adopted it will first be necessary to find the points of cut-off as the valves are now set. As a locomotive engine performs the principal part of its work with the reverse lever hooked back towards the center notch and the valves cutting off at early points in the stroke, it is more im-

portant that the steam should be equally distributed with the lever in the working notch than with it down in the corner.

Passenger engines usually cut-off at from 4 to 6 in., and freight engines at from 6 to 9 in. As in setting the eccentrics, a start at finding the points of cut-off may be made with the engine on either dead center, but for convenience it may be assumed in this instance that the engine has been placed on the right forward dead center. First try the cut-off in backward motion. Pinch the wheels backward until the crosshead has traveled about 6 in. from the extreme travel mark on the front end of the guides. Then stop the motion, and with the point of the valve tram in center C, Fig. 101, move the reverse lever back of the center until the tram will drop into the forward port mark F. Put the lever one notch farther back, then pinch the wheels backward until the tram again drops into the forward port mark F, thus indicating that the point of cut-off has been reached. Now measure the distance from the front travel mark to the front end of the cross head. Suppose it is found to be 7½ in. Chalk this down on the front end of the outside guide. Use the outside guide for the backward motion, because the backward motion eccentric is on the outside. Now pinch the wheels farther back until the steam is cut off on left side back end of cylinder, which can be ascertained by the use of the tram in the same manner as on the right side. It may be assumed that cut-off takes place when the piston has traveled 8¾ in. from beginning of stroke. Now turn the wheels still farther back, until the right pin passes the front center and reaches the point where cut-off takes place, which will be assumed to be 8 in. from commencement of stroke. These figures should all be

marked down with chalk on the outside guides for the backward motion as they are found, and the reverse lever must be left in the same notch until all four points of cut-off for backward motion are located. Next pinch the engine still farther back until the left pin passes the front center and cut-off for this end is reached, which may be taken at 9 in. for the present.

These investigations show that cut-off for the right cylinder occurs at 7½ in. of the backward and 8 in. of the forward stroke, and that for the left cylinder cut-off takes place at 9 in. of the backward and 8¾ in. of the forward stroke. These figures indicate that the right-hand valve is traveling a little too far ahead and the left valve a short distance too far back, and the cut-off for each side may be equalized by slightly changing the lengths of the backward motion eccentric blades; that of the right-hand one must be lengthened and the left one will need to be shortened, and how much to change them may be found as follows:

Taking the left side first, cut-off occurs on the front end of cylinder at 9 in. and on the back end at 8¾ in., and the average is $\frac{8\frac{3}{4}+9}{2} = 8\frac{7}{8}$ in., which is the distance from each end of the stroke at which cut-off will occur when it is equalized. Now pinch the wheels forward enough to bring the crosshead 8⅞ in. from the end of the stroke and enough more to take up all the lost motion when turning back. Next pinch the engine backward until the crosshead is again 8⅞ in. from the beginning of the stroke at the front end, and with the valve tram in center C, Fig. 101, scribe a mark on the valve rod. This mark will be a short distance ahead of center F, and this distance shows how much too far back the valve is

traveling, and the eccentric blade must be shortened enough to throw the valve ahead that much. This will equalize the cut-off for the left side in the backward motion, and the right side should be treated in the same manner, except that in this case the backward motion eccentric blade must be lengthened, because the valve was traveling too far ahead, the cut-off for the forward stroke being 8 in. and for the backward stroke 7½ in., and the average is $\frac{8 + 7\frac{1}{2}}{2} = 7\frac{3}{4}$ in., which will be the point of cut-off for the right side in backward motion when the proper change is made.

This will leave considerable difference between the two sides, the cut-off on the left side occurring at 8⅞ in. and on the right side at 7¾ in., but this will be remedied later on, and the next move will be to equalize the cut-off for the forward motion by commencing with the backward stroke on the right-hand side. Pinch the engine ahead until the pin passes the forward center and draws the crosshead back 6½ in. from the beginning of the stroke. Move the reverse lever ahead to the corner, then move it slowly backward until the valve closes the port, as will be indicated by the valve tram when it reaches from center C to center F, Fig. 101; then put the lever in the first notch ahead of that position and leave it there until the points of cut-off have all been found for the four strokes. Now pinch the engine ahead until the tram again shows that the point of cut-off is reached. This may be assumed to be 8 in. back of beginning of the stroke. Mark this down on the front end of right inside guide, then turn the wheels ahead and get the cut-off for the front end of left-hand cylinder, which will be, say 7 in. Again pinching ahead, find the cut-off

for the back end of the right-hand side to be 8¾ in., and still turning ahead, find cut-off for back end of left side to take place at 8 in.

For convenience, the cut-off for the left side may be equalized first. It was found that cut-off occurred at 7 in. for the backward stroke and at 8 in. for the forward stroke, the average being $\frac{7+8}{2} = 7\frac{1}{2}$ in., and in order to equalize the travel of the valve, which now travels too far ahead, it will be necessary to lengthen the eccentric blade. Pinch the wheels back far enough to bring the crosshead within less than 7½ in. of the end of the stroke, so that when turned ahead again all lost motion may be overcome. Now pinch ahead again until the crosshead is exactly 7½ in. from the beginning of the stroke, and with the valve stem tram in center C scribe a mark on the valve rod, and the distance of this mark from center B is the amount that the eccentric blade must be lengthened. This will equalize the cut-off for forward motion on the left side, and the right side next demands attention. Here the point of cut-off for the backward stroke was 8 in., and for forward stroke 8¾ in., the average being $\frac{8+8\frac{3}{4}}{2} = 8\frac{3}{8}$ in., which is the point at which cut-off for forward motion on the right side must be equalized for the present.

The right forward motion eccentric blades will also need to be lengthened, as the valve travels too far ahead, and the correct amount to lengthen the blade may be found in the same manner as with the left side. This will leave the points of cut-off for forward motion as follows: for right-hand side, 8⅜ in.; left side, 7½ in. In backward motion, as equalized, cut-off for right side

is 7¾ in., left side 8⅞ in. It will thus be seen that in forward gear cut-off is earliest on left side and in back gear it occurs latest on that side. In order to overcome this unequal condition one of two things may be done, either lengthen the link hanger on the left side or shorten hanger on the right side. The former method will be adopted, but before making any alterations it will be necessary to ascertain the amount to lengthen, and this may be done in the following manner:

Put the reverse lever in the same notch that it was in when the cut-off in forward gear was found, and measure the distance from any stationary point directly above or below the upper link hanger pin on the left side to the center of that pin. Now pinch the engine ahead far enough to bring the left crosshead the same distance from the beginning of the stroke as the right crosshead was when cut-off took place, which distance is 8⅜ in. This is where cut-off must occur on the left side also. Now move the reverse lever ahead about four notches, and then with the point of the valve tram in center C move the lever slowly back until cut-off occurs as indicated by the tram. Now measure the distance again, from the same stationary point to the center of the upper hanger pin. The difference between this distance and the distance between these two points as first found is the amount the left hanger must be lengthened to equalize the cut-off on both sides, or raising the tumbling shaft box slightly more than this on the right-hand side would bring about the same result as shortening the link hanger on that side. This change will slightly affect the operation of the valves in back gear, for this reason: The nearer the link block is to the center of the link, the shorter will be the cut-off, and the changes made, viz.,

lengthening the hanger, while it throws the block farther below the center of the link in forward gear, thus delaying the cut-off, at the same time brings the link block nearer the center of the link in back gear, thus accelerating the cut-off, and this is the result wished for to cause the two sides to cut off nearer equal in back gear, as it will be remembered that the cut-off on the left side occurred at 8⅞ in. in the back gear and 7¾ in. on the right side in back gear. The amount that the hanger has been lengthened may not exactly equalize the cut-off in back gear, but it will bring it near enough for all practical purposes, for the reason that the engine does very little work in back gear. Owing to the space occupied by the piston rod in the back end of the cylinder, the cut-off should occur ¼ or ⅜ in. later in the back end than in the front end of the cylinder if it is desired that the same volume of steam be admitted to each end of the cylinder.

The next points to be determined relate exclusively to the exhaust opening and closure with reference to release and compression. These events, as has been already explained, are controlled by inside clearance and inside lap. If a valve is line and line inside, having neither inside clearance nor inside lap, the point M, Fig. 102, will indicate both the opening and closure of the exhaust, but if a valve has inside clearance, release will occur before the valve has reached its central position; or if the valve has inside lap, closure of the exhaust passage will occur before the valve has reached its central position.

Now in order to ascertain at what point in the stroke either one of the above named events takes place, use a pair of small dividers and from center M describe on

each valve rod a small circle, the radius of which equals the inside lap or inside clearance, as the case may be, and make two small centers where the circle crosses the horizontal line, also mark each with some distinguishing mark to show whether it represents inside lap or inside clearance.

Having gotten these marks properly located, proceed to test each event by the same method as with the cut-off, marking down the point in each stroke at which the event, be it release or compression, begins, after which compare the figures, and the changes required may be made in the same way as with the cut-off. Equalizing the cut-off incidentally affects exhaust closure, and as compression is of more importance than release, it should be made as near perfect as possible. There is, however, but one method by which the various events in the working of a valve may be made thoroughly clear, and that is by the use of the indicator.

The maximum port opening and maximum travel of the valve may be found thus: Place the reverse lever in full gear; that is, "down in the corner." Then pinch the wheels one complete revolution, and with the valve tram in center C, mark the extreme travel of the valve in each direction. The distance between the extreme points indicates the maximum travel of the valve, and the distance from either extreme point to the port mark indicates maximum port opening.

The minimum travel and minimum port opening may be found by placing the reverse lever in the center notch of the quadrant, and then repeating the operation of turning the wheels one revolution, while at the same time the distances are noted with the tram in the same manner as before.

DETAILS OF OPERATION OF WALSCHAERT VALVE GEAR.

The operation of the Walschaert valve gear is not disturbed by the vibration, up and down, of the engine upon its springs, to the extent that the Stephenson valve gear is, provided of course that the point of connection of the eccentric rod, and the link is as near the center line of motion of the main rod as it is possible to locate it. Great care should be exercised in adjusting a valve gear to see that the valve is in the center of its travel, when the piston is at mid-stroke and the reverse lever is in the middle or central notch of the quadrant. The position of mid-stroke for the piston does not necessarily imply that the center of the crank pin is perpendicular above or below the center of the driving axle.

On the contrary, owing to the angularity of the main rod, the position of the crank pin, with the piston at mid-stroke, will be a slight distance to one side of a perpendicular line cast through the center of the driving axle, the distance depending upon the length of the rod. The longer the rod the less this distance will be.

The main pin will in this position be on what is termed the working quarter, the center of the crosshead pin will be at the center of its travel or mid-stroke, and the combination lever of the Walschaert valve gear will be in a perpendicular position, which can be ascertained by dropping a plumb line from the center of the connection of valve stem and combination lever, to the center of the connection of crosshead and combination lever.

With the reverse lever in the central notch, the centers of the link fulcrum pin, and link-block pin, should exactly coincide. If they do not it is an indication that the center notch of the quadrant is not properly located, and this defect should be remedied at once before proceeding to test the correctness of the valve's position.

Having finally succeeded in getting all of the other parts correctly located, an off hand test of the valve may be made as follows: If with the cylinder cocks open, the piston at mid-stroke as just described, and reverse lever in center notch, a slight opening of the throttle causes the steam to blow from one of the cylinder cocks on that side of the engine, it is an indication that the valve is not at mid-travel, and that it does not cover the admission ports. Which port is uncovered will be shown by the cylinder cock from which the steam is blowing.

The cause of an error like this in the position of the valve, and the proper remedy for it can only be determined by careful experiment. First it might be corrected by a slight change in the length of the valve stem, but this should be proceeded with very carefully. Secondly, the link bearer, which is usually attached to the guides, may possibly not be in its true position and may require to be removed either back or forward a slight distance.

This latter move is liable to cause another error, however, due to the fact that it changes the distance between the link fulcrum and the eccentric.

But this fault can be easily corrected by adjusting the length of the eccentric rod. Any changes in the Walschaert valve gear, after the engine has been turned out of the back shop, should be proceeded with carefully, always bearing in mind that this gear from the link for-

ward is permanently located, all motion bearers as a rule being attached to the guides. Back of the link, however, slight changes in distances may be made when necessary to take up lost motion due to wear.

The necessity of correcting the length of the eccentric rods may be ascertained by the following method:

Place the engine with one side on the forward dead center, and the reverse lever down in the go-ahead corner notch. Then have a helper slowly move the reverse lever back towards the center notch of the quadrant. Watch the valve stem closely while the link block is rising, and if the stem is pushed forward slightly, it is an indication that the eccentric rod needs to be lengthened.

On the other hand, if the valve stem is drawn backwards by the rising link-block, the eccentric rod should be made slightly shorter. Very slight changes in either case should be made in the length of the rod. It is better to make several tests of hooking up the reverse lever to the center notch, and slightly changing the length of the rod as required at each test. Finally, when, by moving the reverse lever from the corner notch back to the center notch, with the engine in the position as indicated above, neither valve stem nor radius rod is affected, the adjustment for that side of the engine may be considered correct. The same method of testing should be followed on the other side, with the exception that the engine be placed on the back dead center. The facilities for making changes in the length of the eccentric rods on modern locomotives, equipped with the Walschaert gear appear to indicate plainly that it is a "back shop" job in most cases.

One very important feature connected with the successful operation of the Walschaert valve gear, is that

the bracket supporting the link trunnion should be permanently fixed at an unchangeable distance from the cylinder, and valve chest, and in order to accomplish this it is necessary that this bracket be firmly attached to the guide yoke or bearer.

The valve stem slide which supports the combination lever, and forward end of the radius bar, represents another point which permits of no variation in distances, and for this reason it also is securely attached to the guides.

A discussion of the operation of the Walschaert valve gear would not be complete without a reference to the subject of breakdowns and their remedies, and in treating this subject we will start with the eccentric crank, that being the prime mover, and main source of motion for the other parts of the gear. In this connection it may be well to note, particularly, an advantage possessed by this gear over the Stephenson link, viz., that the eccentric is invariably attached to, and derives its motion from the main driving axle, through the medium of the main crank pin. This cannot be said of the Stephenson gear, as on some types of engines the eccentrics are not located on the main axle, but on one of the other axles. Especially is this the case with engines having three or more pairs of driving wheels.

BROKEN SIDE ROD.

The breaking of a side rod on an engine of this type equipped with the Stephenson link would totally disable the engine, for the reason that the driving axle carrying the eccentrics would not remain in accord with the main driving axle, and crank pin still being driven by the main

rod. With the Walschaert valve gear the breaking or removal of a side rod does not totally disable the engine, for the reason as before stated, that the eccentric is actuated by the main crank pin, and the engine on the uninjured side of the locomotive could still be operated.

BROKEN ECCENTRIC OR ROD.

In case an eccentric crank, or eccentric rod of a Walschaert valve gear should break or become disabled in any way, the broken parts should be removed, then lower the link block and radius bar to the bottom of the link, after doing which disconnect the suspension bar from the radius bar, also disconnect the radius bar from the combination lever, and raise the front end of the radius bar clear of interference, and firmly secure it by wiring through the pin hole, and to any convenient and solid part of the engine. This should be done in order to prevent the link from vibrating with the motion of the engine. Having thus secured the link and radius bar, the next move is to block the valve, the methods of doing which will be described later on. Very often a breakdown occurs with some part of the engine that will prevent the working of steam on that side, despite the fact that the valve gear is intact. In such cases it will be necessary to block the valve to prevent admission of steam to the cylinder, while at the same time the other parts of the valve gear need not be wholly taken down or disconnected, excepting of course the radius bar, which should always be disconnected at least, at the front end when it becomes necessary to block the valve.

In case a breakdown of this character should occur, with the engine having but a short distance to travel to

reach its terminal station, a quick method of solving the problem would be to block the valve, disconnect the radius rod at the front end, and suspend the rod by a pendulum of strong wire or rope passed through the pin hole, and attached to some point above in such a manner as to allow the rod to swing freely back and forth with the motion of the link. Care should be exercised to see that the suspended end of the radius bar does not strike anything in its vibrations. However, this method should not be pursued, except as before stated, when the engine has but a short distance to run. A much safer method, and one which will permit the engine to be speeded up, and run for a long distance with one side disabled, would be as follows:

Locate the link block and back end of the radius bar in the exact center of the link by placing the reverse lever in the center notch of the quadrant. Then cut two blocks of wood of the proper dimensions to fit and place them between the lower part of the link and the link block for the purpose of supporting the link block in the center.

These blocks should be secured by lashing with wire or rope. Next disconnect the hanger between the lifting arm, and radius bar, also disconnect the front end of the latter, from the combination lever, and wire the radius bar up as previously directed, or suspend it in any way that will support it securely, there being no motion imparted to it so long as the link block, with the back end of the radius bar, remains at the center of the link.

It is not necessary in all cases to disconnect the combination lever as the motion imparted to its lower end by the crosshead is not likely to disturb the valve, but the precaution should be taken to watch the first move-

ment of the crosshead and see that the combination lever does not strike the crosshead pin, since the motion of the lever may be slightly changed by the disconnection of its upper end.

LINK BLOCK AND RADIUS BAR.

A good plan is to carry wood blocks on the engine of the proper shape and dimensions, to hold the link block, and radius bar at the center of the link. They will then be available at any time that it is desirable to carry the link block and radius bar without motion.

It should be remembered that in this discussion of rules for guidance in case of breakdowns, there is always one feature that must not be neglected, viz., the placing of the valve in the center of its travel in order to prevent admission of steam to the cylinder of the disabled side. This applies always in case of breakdowns when it is desired to proceed with one side disabled. In the case of a broken radius bar, the pieces of the broken bar should all be taken down, except possibly in case there should be enough left of the bar forward of the link to make it safe to block it up in the center of the link as heretofore described, and suspend or secure the front end of the bar.

BROKEN SUSPENSION BAR.

The location of the lifting arm, or suspension bar of the Walschaert valve gear is governed by the type of engine to which it is applied. On some engines it is connected to an extension of the radius bar back of the link, and on others it is connected to the bar forward of the link. In case the lifting arm should

break, or the extension of the radius bar back of the link should become bent or broken from any cause, while the connection of the radius bar to the link block still remained intact, the following method may be pursued:

Assuming that the engine is to run forward, first place the reverse lever in the notch that it will be expected to work in. Then note the height of the link block from the bottom of the link on the side of the engine that is not disabled. Next place a short block in the link under the radius rod, or the link block on the disabled side, this block being of a length sufficient to raise the radius bar and link block to about the same height in the link as the one on the other side is. All broken parts should be removed and another block placed in the link above the link block or radius bar to prevent them from slipping up in the link. Having thus secured all parts, the engine may be operated, always bearing in mind, however, that the reverse lever must not be moved from the notch at first chosen, neither hooked up, nor dropped lower, so long as the engine is working with one of the lifters disabled.

BROKEN VALVE STEM.

When a valve stem breaks, disconnect the radius bar and secure it in the manner already described. Then place the valve in its central position and block or clamp it there. The combination lever may be left in place, provided that its lower portion does not strike the crosshead pin in its travels. As has been stated, the motion of the combination lever is liable to be changed slightly, due to its having been disconnected at the top from the radius bar, but in case of a broken valve stem, the slide from which the valve stem has been disconnected by the

breakage, and to which the combination lever is still connected, may now be placed, and secured in any position on the slide bar, that offers the safest motion to the combination lever, and it should then be blocked in that position in order to preclude any possible chance of its being moved by the combination lever to such an extent as to affect the valve in the least, or push it off center.

BROKEN COMBINATION LEVER.

In case the combination lever, or the short bar connecting it to the crosshead should get broken or bent, the pieces should be removed, and the radius bar disconnected, and dealt with in the manner already explained, after which the valve should be placed at mid travel and blocked.

BLOCKING THE VALVE.

When an engine that is equipped with the Walschaert valve gear is disabled on account of a broken main rod, and no other parts of the engine have been disabled or broken, one of the first things to do is to disconnect the radius rod and secure it in the manner heretofore outlined. Then if the valve is of the inside admission type, it should be pushed to the forward end of the valve chest and secured by a clamp on the stem, or it may be blocked at the slide, to hold it in that position.

If the valve is an outside admission valve it should be secured at the back end of the valve chest in such a manner as to prevent it from moving. The object aimed at in securing the valve, in either case, is to keep open the admission port for the front end of the cylinder, in order to permit the steam to enter the cylinder through that port, while the back port is kept open to the exhaust.

BLOCKING THE CROSSHEAD.

The next move is to draw the crosshead towards the back end of the guides as far as it will go or until the piston is up snugly against the back cylinder head, and there block the crosshead. This will hold the piston firmly in that position, whether the throttle is open or closed, and there will be no danger of its moving.

Of course, when the throttle is open, the piston will be blocked by the steam admitted through the front port, but in drifting down hill, or if the engine should receive a sudden jerk caused by an application of the air brakes, or from any other cause, there is danger of the piston slamming up against the front cylinder head, unless the crosshead is securely blocked.

The combination lever need not be disturbed. If the eccentric crank and rod have not been damaged in any way they may also be left in position, as the motion imparted to the link by the eccentric rod will do no harm, so long as the radius bar is properly suspended.

Having removed the parts of the broken main rod, and attended to the above described details, the engine is ready to proceed, with one side in working condition.

BROKEN PISTON OR PISTON ROD.

If a piston gets broken, or disconnected from the piston rod, the usual result is that the front cylinder head also gets broken Whether the front head is broken or not it should always be removed, and the broken piston taken out of the cylinder. If the piston rod is not broken the main rod may be left up, but the radius bar must be disconnected, and secured, and the valve

centered and blocked. The engine is now ready to go, provided the piston rod has not been bent, but if the rod is bent in the least, the main rod should be taken down also, and the crosshead be blocked in the manner already recommended.

BROKEN CYLINDER HEAD.

In case a front cylinder head is broken, or blown out, the radius bar should be disconnected, as usual, and the valve centered. The back cylinder cock should be removed, also, in order to prevent the formation of a vacuum behind the piston in its travels, and also to allow any water of condensation to escape. The lubrication of the piston with the front cylinder head removed becomes an easy problem.

If the back cylinder head should get broken or blown out, an accident which fortunately does not occur very often, the main rod should be taken down, as it would not be safe to take any chances on the piston rod having been bent when the cylinder head was broken.

If the breakdown is not serious, as for instance, a piece broken out, but still leaving the head strong enough to withstand the pressure of the piston against it, while under steam, the same method may be pursued as in the case of a broken main rod, viz., blocking the piston with steam admitted through the front port, also blocking the crosshead.

In case the back cylinder head is broken very badly, in fact rendered entirely useless except for scrap, the main rod should be taken down, the radius bar disconnected, and secured as already explained, and the valve placed at mid travel, and blocked.

BROKEN CONNECTING BAR, FOR COMBINATION LEVER.

If at any time it should happen that the short connecting bar between the crosshead and the lower end of the combination lever, should get broken, and lost, from both sides of the engine, and it is desired to obtain the lengths of the lost bars, the following method may be resorted to, and it will give at least a close approximation:

Place the crosshead at mid travel, in that position which would indicate that the piston is at mid stroke. Then place the reverse lever in the center notch of the quadrant, which should bring the link block pin and the link fulcrum pin or trunnion center to center.

When the above mentioned parts are in the positions designated, viz., the piston at mid stroke, and the link block, and link trunnion centers together, the long arm of the combination lever should hang perpendicular, and it may be placed in this position by the use of a plumb line, after which the length of the lost bar may be obtained by taking the distance from the center of the pin in combination lever to the center of the pin in the arm on the crosshead, to which the short bar is connected.

QUESTIONS.

311. What does the correct setting of the valves of a locomotive mean?

312. What two very important details should be looked after first when preparing to set valves?

313. What should be done regarding the eccentric rods?

314. With indirect valve gear, what is the position of the eccentric that controls the valve?

315. What is meant by angular advance of the eccentrics?

316. What should be done with the reverse lever before commencing to set valves?

317. What is the next most important proceeding in valve setting?

318. How should the valve rod connect with the rocker arm?

319. If the valve rod should be cramped or twisted, how would this affect the valve?

320. What should be done with the steam chest and valve stem gland?

321. What should be done with the lost motion between the valve and valve yoke while getting the port marks?

322. Where should the port marks be placed for convenience?

323. Where will the point indicating mid travel or central position be?

324. How is the lap indicated by the marks that are now on the valve stem?

325. If the valve has neither inside lap nor inside clearance, what point indicates release and compression?

326. If the valve has inside lap or inside clearance, how may they be measured and properly marked on the valve stem?

327. What is the next important move in valve setting after getting the port marks?

328. What is the meaning of the term dead center as applied to an engine?

329. How many dead centers must the crankpin pass in each revolution?

330. How many dead centers are to be located and marked in setting the valves of a locomotive?

331. Describe in general terms the method of locating and marking a dead center.

332. How should the guides be marked while the engine is on dead center?

333. How are the marks for lap or lead located on the valve rod?

334. What should be done before making any adjustments?

335. When marks for lap or lead are located on the valve stem, how are they distinguished from each other?

336. How are the eccentric rods adjusted as to length?

337. Give the rule for finding out how much the eccentric blade must be lengthened or shortened in order to get the correct travel for the valve.

338. Suppose the valve has ¼-in. lead on the forward port and 1-16-in. lap on the back port, how much must the blade be shortened?

339. If a valve has 1-16-in. lead at the forward port and ⅛-in. lap on the back port, what should be done with the eccentric rod?

340. After the lengths of all the eccentric blades have been adjusted so as to give the valve correct travel, what comes next?

341. Which eccentric should be set first?

342. Why?

343. What precaution should be taken when turning the eccentrics ahead to increase the lead?

344. After getting the eccentrics in their correct po-

sition and firmly secured, what is the next move in valve setting?

345. Mention some of the causes for variation in the cut-off.

346. How may this variation in the cut-off be equalized?

347. Mention another very common method of equalizing the cut-off.

348. Where should the reverse lever be placed while equalizing the cut-off?

349. At what point in the stroke do passenger engines usually cut-off?

350. At what point in the stroke do freight engines usually cut-off?

351. By what means may the point of cut-off be ascertained?

352. Suppose it is found that for the left-hand cylinder cut-off occurs at 9 in. of the backward and 8¾ in. of the forward stroke, at what distance from each end should it occur when equalized?

353. What must be done to bring about this equalization?

354. If, after equalizing the cut-off on both sides, it is found that on the right-hand side it occurs at 8⅜ in. in forward gear and 7¾ in. in back gear, and that on the left side cut-off occurs at 7½ in. in forward and 8⅞ in. in back gear, what may be done to overcome the difficulty?

355. How may the amount to lengthen or shorten the link hanger be ascertained?

356. Why should cut-off occur a little later in the back end than in the front end of the cylinder?

357. What are the next points to be determined?

358. If a valve has inside lead or clearance, how will release be affected?

359. How may the points of release and compression be ascertained?

360. How does equalizing the cut-off affect exhaust closure?

361. Which is the more important, release or compression?

362. How may the maximum port opening and maximum travel of the valve be found?

363. How may the minimum port opening and minimum valve travel be found?

THE INDICATOR.

The Indicator. One of the greatest aids to the economical operation of the steam engine is the indicator, and it is the privilege of every engineer to have at least an elementary, if not a thorough knowledge of its principles and working. The time devoted to the study of the indicator, and in its application to the engine, is time well spent, and in the end will well repay the student of steam engineering.

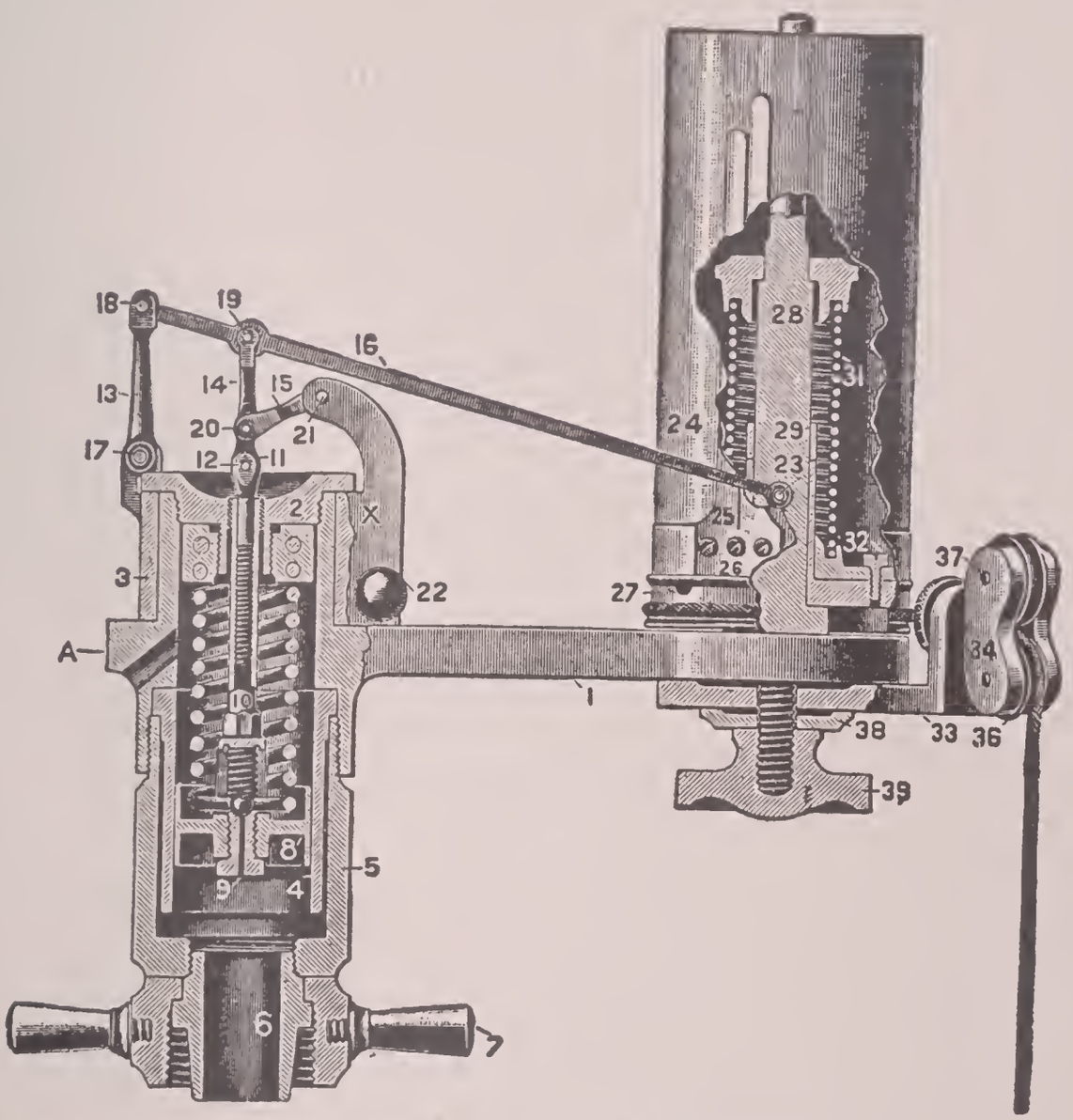
Inventor. The indicator was invented and first applied to the steam engine by James Watt, whose restless genius was not satisfied with a mere outside view of his engine as it was running, but he desired to know more about the action of the steam in the cylinder, its pressure at different portions of the stroke, the laws governing its expansion after being cut off, etc. Watt's indicator, although crude in its design and construction, contained embodied within it all of the principles of the modern instrument.

Principles. These principles are:

First. The pressure of the steam in the engine cylinder throughout an entire revolution, against a small piston in the cylinder of the indicator, which in turn is controlled or resisted in its movement by a spring of known tension, so as to confine the stroke of the indicator piston within a certain small limit.

Second. The stroke of the indicator piston is communicated by a multiplying mechanism of levers and parallel motion, to a pencil moving in a straight line; the distance through which the pencil moves being governed

by the pressure in the engine cylinder and the tension of the spring.



SECTIONAL VIEW CROSBY INDICATOR

Fig. 108.

Third. By the intervention of a reducing mechanism and a strong cord, the motion of the piston of the engine throughout an entire revolution is communicated to a small drum attached to and forming a part of the indicator. The movement of the drum is rotative and in a direction at right angles to the movement of the pencil.

The forward stroke of the engine piston causes the drum to rotate through part of a revolution and at the same time a clock spring connected within the drum is wound up. On the return stroke the motion of the drum is reversed, and the tension of the spring returns the drum to its original position, and also keeps the cord taut.

To the outside of the drum a piece of blank paper of suitable size is attached and held in place by two clips. Upon this paper the pencil in its motion up and down traces a complete diagram of the pressures, and other interesting events transpiring within the engine cylinder during the revolution of the engine. In fact, the diagram traced upon the paper is the compound result of two concurrent movements. First, that of the pencil, caused by the pressure of the steam against the indicator piston; second, that of the paper drum, caused by, and coincident with the motion of the engine piston. The upper end of the indicator cylinder is always open to the atmosphere, the steam acting only upon the under side of the small piston, and when the cock connecting the cylinders of the engine and indicator is closed, both ends of the indicator cylinder are open to atmosphere pressure, and the pencil then stands at its neutral position. If now the pencil is held against the paper and the drum rotated either by hand, or by connecting it with a cord, a horizontal line will be traced. This line is called the atmospheric line, meaning the line of atmospheric pressure, and it is a very important factor in the study of the diagram.

Crosby Indicator Spring
Fig. 109.

On a locomotive, the pencil, in tracing the diagram, will not, or at least should not, fall below the atmospheric line at any point, but will on the return stroke trace a line called the line of back pressure.

As before stated, the length of stroke of the indicator piston, and the pencil movement as well, is controlled by a spiral steel spring which acts in resistance to the pressure of the steam. These springs are made of different tensions, in order to be suitable to different steam pressures and speeds, and are numbered 20, 40, 60, etc., the number meaning that a pressure per square inch in the engine cylinder corresponding to the number on the spring will cause a vertical movement of the pencil through a distance of one inch. Thus, if a number 20 spring is used and the pressure in the cylinder at the commencement of the stroke is 20 lbs. per square inch, the pencil will be raised one inch, or if the pressure is 30 lbs., the pencil will travel 1½ in., and if there is a vacuum of 20 in. in the condenser, the pencil will drop ½ in. below the atmospheric line, for the reason that 20 in. of vacuum corresponds to a pressure of about 10 lbs. less than atmospheric pressure or an absolute pressure of about 4 lbs. If a 60 spring is used, a pressure of 60 lbs. in the engine cylinder will be required to raise the pencil one inch, or 90 lbs. to raise it 1½ in.

The Ashcroft Manufacturing Co. of New York, makers of the well known Tabor indicator, have recently introduced a new feature in indicator work by connecting the spring on top of the cylinder and in plain view of the operator. This arrangement removes the spring from the influence of direct contact with the steam, and it is subject only to the temperature of the surrounding atmosphere. It is claimed that as a result of this the ac-

curacy of the spring is insured and that no allowance need be made in its manufacture for expansion caused by the high temperature to which it is subject when

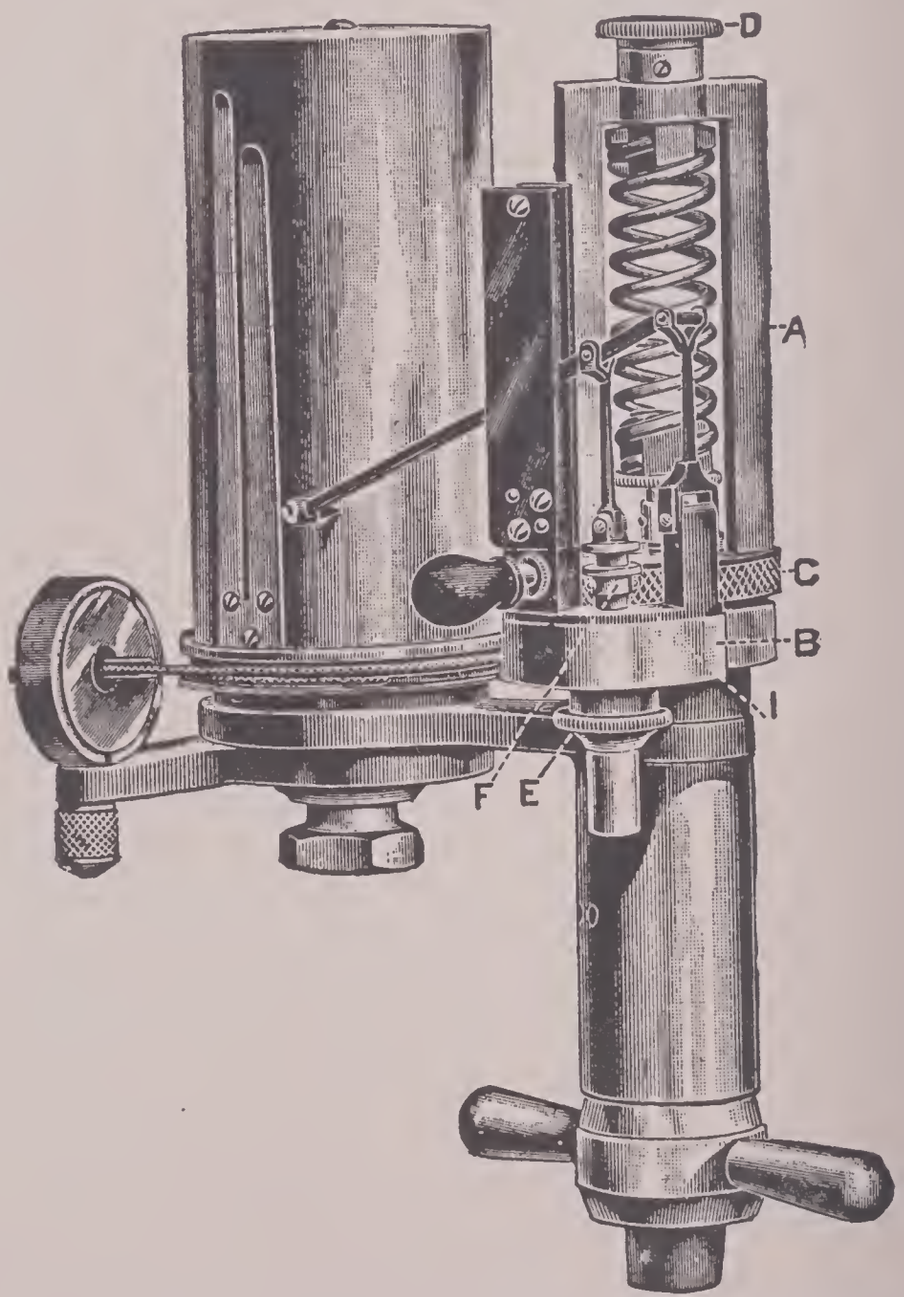


Fig. 110. Improved Tabor Indicator with Outside Connected Spring Ashcroft Mfg. Co., N. Y.

located within the cylinder. Another good feature of this design is, that the spring can be easily removed without disconnecting any one part of the instrument in

case it is desired to change springs. A cut of the improved instrument is herewith presented.

Fig. 111 is a sectional view of the American Thompson improved indicator. Fig. 112 shows the spring. Fig. 113 is a three-way cock for attaching the indicator to the cylinder.

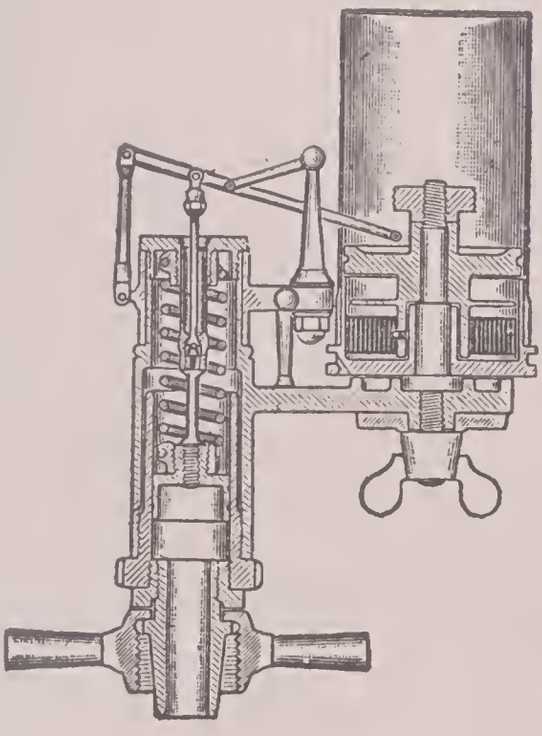


Fig. 111.

Fig. 112.

Reducing Mechanism. Probably the only practically universal mechanism for reducing the motion of the crosshead is a reducing wheel, a device in which, by the employment of gears and pulleys of different diameters, the motion is reduced to within the compass of the drum, and the device is applicable to almost any make of engine, whether of high or low speed. Some makers of indicators attach the reducing wheel directly to the indicator, thus producing a neat and very convenient arrangement. Fig. 114 shows the indicator complete, with reducing wheel attached.

Attaching the Indicator. The cylinders of most engines at the present time are drilled and tapped for indicator connections before they leave the shop, which is eminently proper, as no engine builder, or purchaser either, should be satisfied with the performance of a new engine until after it has been accurately tested and adjusted with the indicator.

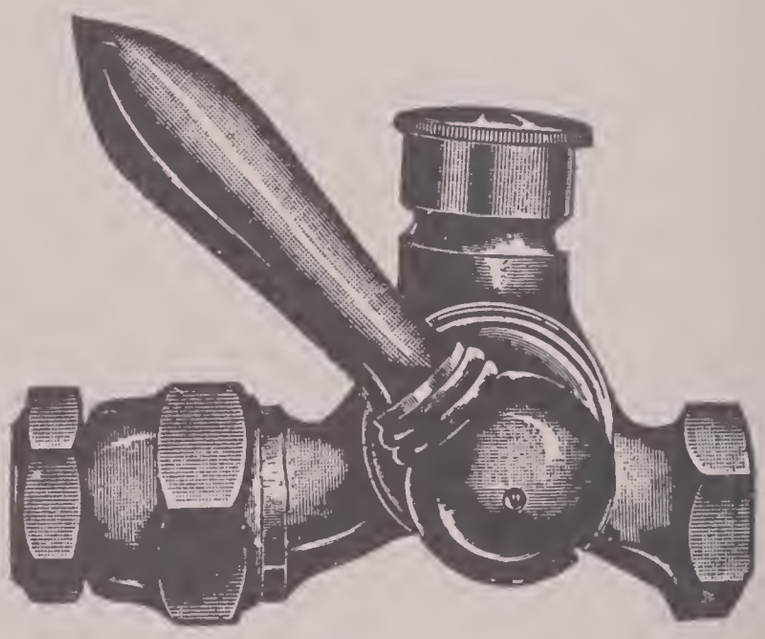


Fig. 113.

The main requirements in these connections are that the holes shall not be drilled near the bottom of the cylinder where water is likely to find its way into the pipes, neither should they be in a location where the inrush of steam from the ports will strike them directly, nor where the edge of the piston is liable to partly cover them when at its extreme travel. An engineer before he undertakes to indicate an engine should satisfy himself that all these requirements are fulfilled. Otherwise he is not likely to obtain a true diagram. The cock supplied with the indicator is threaded for one-half inch pipe, and unless the engine has a very long stroke it is the

practice to bring the two end connections together at the side or top of the cylinder and at or near the middle of its length, where they can be connected to a three-way cock. The pipe connections should be as short and as free from elbows as possible, in order that the steam may strike the indicator piston as nearly as possible at the same moment that it acts upon the engine piston.

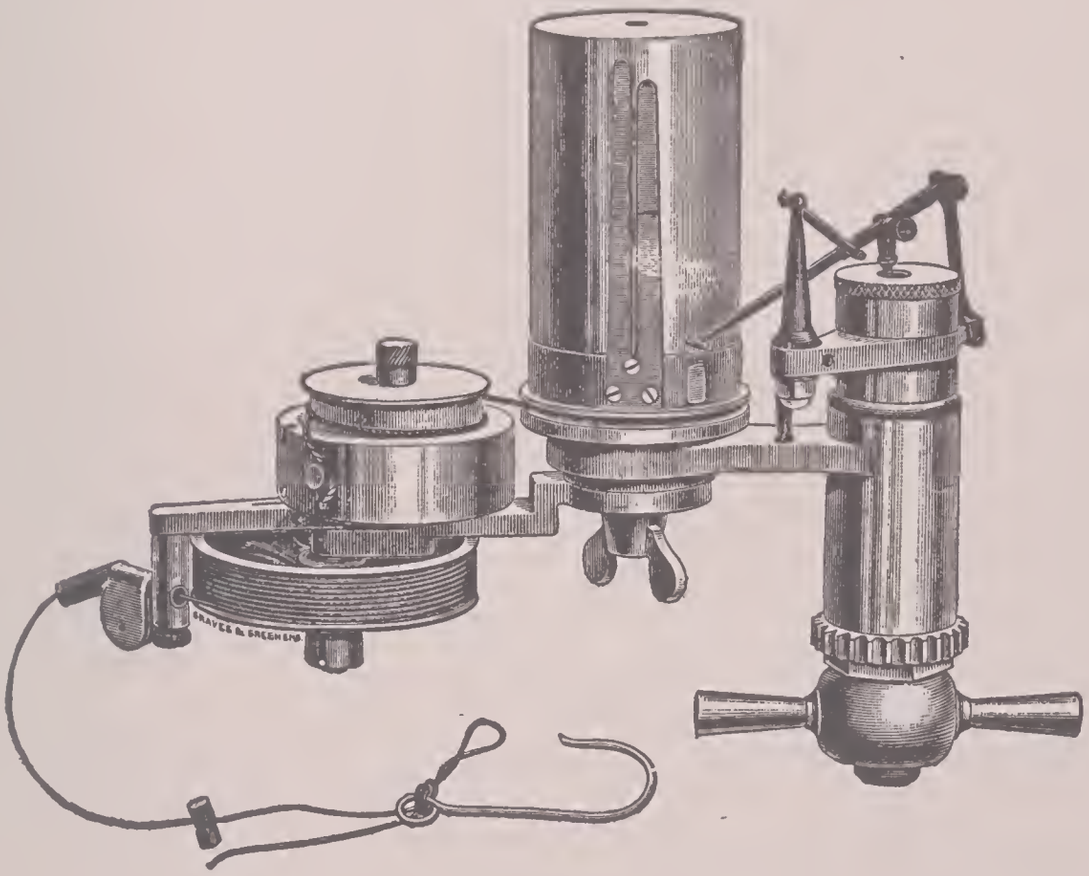


Fig. 114.

These pipes should always be thoroughly blown out and cleaned, by allowing the steam to blow through the open three-way cock during several revolutions of the engine, before connecting the indicator. If this is not done there is a moral certainty that dirt and grit will get into the cylinder of the indicator and cause it to work badly, and give diagrams that are misleading. As

before stated, the height of the diagram depends upon the tension or number of the spring. It is a convenient practice to select a spring numbered one-half of the boiler pressure, as, for instance, suppose gauge pressure or boiler pressure is 200 lbs. per sq. in., then a 100 spring would give a diagram 2 inches in height, which is a convenient height. As to the length of the diagram, this is regulated by adjustment of the cord in its travel, by means of the reducing wheel. Any length of diagram up to four inches may be obtained, but two and a half to three inches is a very good length for analysis.

Care of the Instrument. The indicator should be cleaned and oiled both before and after using. The best material for wiping it is a clean piece of old soft muslin of fine texture, as there is not so much liability of lint sticking to, or getting into, the small joints.

Good clock oil should be used for the joints and springs, and just before taking diagrams it is a good practice to rub a small portion of cylinder oil on the piston and on the inside of the cylinder, but when about to put the instrument away, these should be cleaned and oiled with clock oil also. None but the best cord should be used for connecting the reducing wheel with the crosshead, as a cord that is liable to stretch will cause trouble. Suitable cord, and also blank diagrams, can generally be obtained from firms engaged in manufacturing and selling indicators. After the indicator has been screwed on to the cock connecting with the pipe, the cord must be adjusted to the proper length before hooking it on to the drum. This must be done while the engine is running, by taking hold of the loop on the cord connected with the crosshead with one hand, and with the other hand grasp the hook on the cord attached to the reducing

wheel; then, by holding the two ends near each other during a revolution or two of the crank pin, it will be seen whether the long cord needs to be lengthened or shortened. Care should be exercised in placing the paper on the drum to see that it is stretched tight and firmly held by the clips. The pencil point, having been first sharpened by rubbing it on a piece of fine emery cloth or sandpaper, should be adjusted by means of the pencil stop with which all indicators should be provided, so that it will have just sufficient bearing against the paper to make a fine, plain mark. If the pencil bears too hard on the paper it will cause unnecessary friction and the diagram will be distorted. The best method of ascertaining this fact, and also, whether the travel of the drum is equally divided between the stops, is to place a blank diagram on the drum, connect the cord and while the engine makes a revolution hold the pencil against the paper. Then unhook the cord, remove the paper and if the travel of the drum is not divided correctly it can be changed.

Having thus arranged all the preliminary details, place a fresh blank on the drum, being careful to keep the pencil out of contact with it, connect the cord, open the cock admitting steam to the indicator, and after the pencil has made a few strokes to allow the cylinder to become warmed up, then gently swing it around to the paper drum and hold it there while the engine makes a complete revolution. Then move the pencil clear of the paper, close the cock and unhook the cord. Now trace the atmospheric line by holding the pencil against the paper while the drum is revolved by hand. This method of tracing the atmospheric line is preferable to that of tracing it immediately after closing the cock and while

the drum is still being moved by the engine, for the reason that there is not so much liability of getting the atmospheric line too high owing to the presence of a slight pressure of steam remaining under the indicator piston for a second or two just after closing the cock; also the line drawn by hand will be longer than one drawn while the drum is moved by the motion of the engine and will therefore be more readily distinguished from the line of back pressure.

Having secured a truthful diagram, it now remains to take as many as are desired, and they should follow each other as rapidly as possible, in order that each pair of diagrams may be taken under the same conditions of initial pressure, cut-off, etc. In order to get accurate results, the operator should have an assistant posted in the cab, whose duty it will be to watch the steam gauge, and see that other conditions are the same at least during the time a pair of cards is being taken. As soon as the diagrams are taken, the following data should be noted upon them: the end of cylinder, whether head end, or crank end, boiler pressure, revolutions per minute, miles per hour, throttle opening, cut off. Other data, such as mean effective pressure, back pressure, indicated horse-power, and steam per indicated horse-power, per hour, may be ascertained by an analysis of the diagrams, and should also be noted upon the back of each pair of diagrams after they have been found by calculation. The diagrams should be numbered, also, as they are taken.

The taking of indicator diagrams from locomotives has of late years been greatly facilitated by the use of electrical apparatus whereby any number of diagrams may be taken simultaneously. This is certainly a great

improvement over the old method of hand manipulation, especially for high speed engines.

In order to facilitate the study and analysis of indicator diagrams, the following definitions of technical terms, some of which have already been explained in another part of this book, are here given:

Absolute Pressure. Pressure reckoned from a perfect vacuum. It equals the boiler pressure plus the atmospheric pressure.

Boiler Pressure or Gauge Pressure. Pressure above the atmospheric pressure as shown by the steam gauge.

Initial Pressure. Pressure in the cylinder at the beginning of the stroke.

Terminal Pressure (*T. P.*). The pressure that would exist in the cylinder at the end of the stroke provided the exhaust valve did not open until the stroke was entirely completed. It may be graphically illustrated on the diagram by extending the expansion curve by hand to the end of the stroke. It is found theoretically by dividing the pressure at point of cut-off by the ratio of expansion. Thus, absolute pressure at cut-off = 100 lbs., ratio of expansion = 5; then 100 ÷ 5 = 20 lbs., absolute terminal pressure.

Mean Effective Pressure (*M. E. P.*). The average pressure acting upon the piston throughout the stroke minus the back pressure.

Back Pressure. Pressure which tends to retard the forward stroke of the piston. Indicated on the diagram from a non-condensing engine by the height of the back pressure line above the atmospheric line. In a condensing engine the degree of back pressure is shown by the height of the back pressure line above an imaginary line representing the pressure in the condenser correspond-

ing to the degree of vacuum in inches, as shown by the vacuum gauge.

Total or Absolute Pressure, in either a condensing or non-condensing engine, is that indicated on the diagram by the height of the line of back pressure above the line of perfect vacuum.

Ratio of Expansion. The proportion that the volume of steam in the cylinder at point of release bears to the volume at cut-off. Thus, if the point of cut-off is at one-fifth of the stroke, and release does not take place until the end of the stroke, the ratio of expansion, or in other words, the number of expansions is 5. When the T. P. is known the ratio of expansion may be found by dividing the initial pressure by the T. P.

Wire-Drawing. When through insufficiency of valve opening, or contracted ports, the steam is prevented from following up the piston at full initial pressure until the point of cut-off is reached, it is said to be wire-drawn. It is indicated on the diagram by a gradual inclination downwards of the steam line from the admission line to the point of cut-off. Too small a steam pipe from boiler to engine will also cause wire-drawing and fall of pressure.

Condenser Pressure may be defined as the pressure existing in the condenser of an engine, caused by the lack of a perfect vacuum; as, for instance, with a vacuum of 25 in. there will still remain the pressure due to the 5 in. which is lacking. This will be about 2.5 lbs.

Vacuum. That condition existing within a closed vessel from which all matter, including air has been expelled. It is measured by inches in a column of mercury contained within a glass tube a little over 30 in. in height, having its lower end opened and immersed in a small

vessel filled with mercury. The upper end of the glass tube is connected with the vessel in which the vacuum is to be produced. When no vacuum exists the mercury will leave the tube and fill the lower vessel. When a vacuum is maintained in the condenser, or other vessel, the mercury will rise in the glass tube to a height corresponding to the degree of vacuum. If the mercury rises to the height of 30 in. it indicates a perfect vacuum, which means the absence of all pressure within the vessel, but this condition is never realized in practice; the nearest approach to it being about 28 in.

For the purposes of convenience the mercurial vacuum gauge is not generally used, it having been replaced by the Bourdon spring gauge, although the mercury gauge is used for testing.

The vacuum in a condenser is generally maintained by an air pump, although it can be produced and maintained by the mere condensation of the steam as it enters the condenser by allowing a spray of cold water to strike it. The steam when it first enters the condenser drives out the air and the vessel is filled with steam, which, when condensed, occupies about 1,600 times less space than it did before being condensed; hence a partial vacuum is produced.

While the vacuum in a condenser cannot be considered as power at all, yet it occupies the anomalous position of increasing, by its presence, the capacity of the engine for doing work. This is owing to the fact that the atmospheric pressure or resistance which is always ahead of the piston in a non-condensing engine is, in the case of a condensing engine, removed to a degree corresponding to the height of the vacuum, thus making available just so much more of the pressure behind the

piston. Thus, if the average steam pressure throughout the stroke is 30 lbs. and there is a vacuum of 26 in. maintained in the condenser, there will be 13 lbs. of resistance per square inch removed from in front of the piston, thus making available 30 + 13 = 43 lbs. per square inch.

Absolute Zero has been fixed by calculation at 461.2° below the zero of the Fahrenheit scale.

Piston Displacement. The space or volume swept through by the piston in a single stroke. Found by multiplying the area of piston by length of stroke.

Piston Clearance. The distance between the piston and cylinder head when the piston is at the end of the stroke.

Steam Clearance, Ordinarily Termed Clearance. The space between the piston at the end of the stroke and the valve face. It is reckoned in per cent of the total piston displacement.

Horse-Power (H. P.). 33,000 pounds raised one foot high in one minute of time.

Indicated Horse-Power (I. H. P.). The horse-power as shown by the indicated diagram. It is found as follows:

Area of piston in square inches × M. E. P. × piston speed in feet ÷ 33,000.

Piston Speed. The distance in feet traveled by the piston in one minute. It is the product of twice the length of stroke expressed in feet multiplied by the number of revolutions per minute.

R. P. M. Revolutions per minute.

Net Horse-Power. I. H. P. minus the friction of the engine.

Compression. The action of the piston as it nears the

end of the stroke, in reducing the volume and raising the pressure of the steam retained in the cylinder ahead of the piston by the closing of the exhaust valve.

Boyle's or Mariotte's Law of Expanding Gases. "The pressure of a gas at a constant temperature varies inversely as the space it occupies." Thus, if a given volume of gas is confined at a pressure of 50 lbs. per square inch and it is allowed to expand to twice its volume, the pressure will fall to 25 lbs. per square inch.

Adiabatic Curve. A curve representing the expansion of a gas which loses no heat while expanding. Sometimes called the curve of no transmission.

Isothermal Curve. A curve representing the expansion of a gas having a constant temperature, but partially influenced by moisture, causing a variation in pressure according to the degree of moisture or saturation. It is also called the theoretical expansion curve.

Expansion Curve. The curve traced upon the diagram by the indicator pencil, showing the actual expansion of the steam in the cylinder.

First Law of Thermodynamics. Heat and mechanical energy are mutually convertible.

Power. The rate of doing work, or the number of foot-pounds exerted in a given time.

Unit of Work. The foot-pound, or the raising of one pound weight one foot high.

First Law of Motion. All bodies continue either in a state of rest, or of uniform motion in a straight line, except in so far as they may be compelled by impressed forces to change that state.

Work. Mechanical force or pressure cannot be considered as work unless it is exerted upon a body and causes that body to move through space. The product

of the pressure multiplied by the distance passed through and the time thus occupied is work.

Momentum. Force possessed by bodies in motion, or the product of mass and density.

Dynamics. The science of moving powers, or of matter in motion, or of the motion of bodies that mutually act upon each other.

Force. That which alters the motion of a body, or puts in motion a body that was at rest.

Maximum Theoretical Duty of Steam is the product of the mechanical equivalent of heat, viz., 778 ft. lbs., multiplied by the total heat units in a pound of steam. Thus, in one pound of steam at 212° reckoned from 32° the total heat equals 1,146.6 heat units. Then 778 × 1,146.6 equals 892,054.8 ft. lbs. = maximum duty.

Steam Efficiency may be expressed as follows: $\frac{\text{Heat converted into useful work}}{\text{Heat expended}}$ and maximum efficiency can only be attained by using steam at as high an initial pressure as is consistent with safety and at as large a ratio of expansion as possible. The percentage of efficiency of steam used at atmospheric pressure in a non-expansive engine is very low; as, for instance, the heat expended in the evaporation of one pound of water at 32° into steam at atmospheric pressure = 1,146.6 heat units, and the volume of steam so generated = 26.36 cu. ft.

One cubic foot of steam at 212° contains energy equal to 144 × 14.7 = 2,116.8 ft. lbs., and 26.36 cu. ft. = 2,116.8 × 26.36 = 55,798.84 ft. lbs., which divided by the mechanical equivalent of heat, viz., 778 ft. lbs. = 71.72 heat units, available for useful work. The per cent of

efficiency, therefore, is $\frac{71.72 \times 100}{1,146.6} = 6.28$ per cent. But, suppose the initial pressure to have been 200 lbs. absolute, and that the steam is allowed to expand to thirty times its original volume. The heat expended in evaporating a pound of water at 32° into steam at 200 lbs. absolute pressure = 1,198.3 heat units, and the volume of steam so generated = 2.27 cu. ft. The average pressure during expansion would be 29.34 lbs. per square inch and the volume when expanded thirty times would equal $2.27 \times 30 = 68.1$ cu. ft.

One cubic foot of steam at 29.34 lbs. pressure equals $144 \times 29.34 = 4,224.96$ ft. lbs., and 68.1 cu. ft. will equal $4,224.96 \times 68.1 = 287,719.7$ ft. lbs. of energy, which divided by the equivalent, 778, equals 370.2 heat units, and the per cent of efficiency will be $\frac{370.2 \times 100}{1,198.3} = 30.8$ per cent.

Engine Efficiency. If the engine is considered merely as a machine for converting into useful work the heat energy in the steam regardless of the cost of fuel, its efficiency may be expressed as follows:

$$\frac{\text{Heat converted into useful work}}{\text{Total heat received in the steam}}$$

Example. Assume an engine to be receiving steam at 95 lbs. absolute pressure, that the consumption of dry steam per horse-power per hour equals 20 lbs., that the friction of the engine amounts to 15 per cent, and that the temperature of the feed water is raised from 60° to 170° by utilizing a portion of the exhaust.

In a pound of steam at 95 lbs. absolute there are 1,180.7 heat units, and in a pound of water at 170° there are 138.6 units of heat, but 28.01 of these heat units

were in the water at its initial temperature of 60°. Therefore the total heat added to the water by the exhaust steam equals 138.6 — 28.01 = 110.59 heat units, and the total heat in each pound of steam to be charged up to the engine is 1,180.7 — 110.59 = 1,070.11, and the total for each horse-power developed per hour will be 1,070.11 × 20 = 21,402.2 heat units.

A horse-power equals 33,000 ft. lbs. per minute, or sixty times 33,000 = 1,980,000 ft. lbs. per hour. From this must be deducted 15 per cent for friction of the engine, leaving 1,683,000 ft. lbs. for useful work. Dividing this by the equivalent, viz., 778 ft. lbs., gives 2,163.2 heat units as the heat converted into one horse-power of work in one hour, and the percentage of efficiency of the engine will be $\frac{2,163.2 \times 100}{21,402.2} = 10.1$ per cent.

Efficiency of the Plant as a Whole includes boiler and engine efficiency and is to be figured upon the basis of

$$\frac{\text{Heat converted into useful work}}{\text{Calorific or heat value of fuel.}}$$

Horse-Power Constant of an engine is found by multiplying the area of the piston in square inches by the speed of the piston in feet per minute, and dividing the product by 33,000. It is the power the engine would develop with one pound mean effective pressure. To find the horse-power of the engine, multiply the M. E. P. of the diagram by this constant.

Logarithms. A series of numbers having a certain relation to the series of natural numbers, by means of which many arithmetical operations are made comparatively easy. The nature of the relation will be understood by considering two simple series, such as the fol-

lowing, one proceeding from unity in geometrical progression and the other from 0 in arithmetical progression:

Geom. series, 1, 2, 4, 8, 16, 32, 64, 128, 256, 512, etc.

Arith. series, 0, 1, 2, 3, 4, 5, 6, 7, 8, 9, etc.

Here the ratio of the geometrical series is 2 and any term in the arithmetical series expresses how often 2 has been multiplied into 1 to produce the corresponding term of the geometrical series. Thus, in proceeding from 1 to 32 there have been 5 steps or multiplications by the ratio 2; in other words, the ratio of 32 to 1 is compounded 5 times of the ratio of 2 to 1. The above is the basic principle upon which common logarithms are computed.

Hyperbolic Logarithms. Used in figuring the M. E. P. of a diagram from the ratio of expansion and the initial pressure. Thus, hyperbolic logarithm of ratio of expansion + 1 multiplied by absolute initial pressure and divided by ratio of expansion = mean forward pressure. From this deduct total back pressure and the remainder will be mean effective pressure. The hyperbolic logarithm is found by multiplying the common logarithm by the constant 2.302585. Table 50a gives the hyperbolic logarithms of numbers usually required in calculations of the above nature.

Steam Consumption per Horse-Power per Hour. The weight in pounds of steam exhausted into the atmosphere or into the condenser in one hour divided by the horse-power developed. It is determined from the diagram by selecting a point in the expansion curve just previous to the opening of the exhaust valve and measuring the absolute pressure at that point. Then the piston displacement up to the point selected, plus the

Table 51.—Hyperbolic Logarithms.

No.	Log.	No.	Log.	No.	Log.	No.	Log.	No.	Log.
1.01	0.0099	3.00	1.0986	5.00	1.6094	7.00	1.9459	9.00	2.1972
1.05	0.0487	3.05	1.1151	5.05	1.6194	7.05	1.9530	9.05	2.2028
1.10	0.0953	3.10	1.1341	5.10	1.6292	7.10	1.9600	9.10	2.2083
1.15	0.1397	3.15	1.1474	5.15	1.6390	7.15	1.9671	9.15	2.2137
1.20	0.1823	3.20	1.1631	5.20	1.6486	7.20	1.9740	9 20	2.2192
1.25	0.2231	3.25	1.1786	5.25	1.6582	7.25	1.9810	9.25	2.2246
1.30	0.2623	3.30	1.1939	5.30	1.6677	7.30	1.9879	9.30	2.2310
1.35	0.3001	3.35	1.2090	5.35	1.6771	7.35	1.9947	9.35	2.2354
1.40	0.3364	3.40	1.2238	5.40	1.6864	7.40	2.0015	9.40	2.2407
1.45	0.3715	3.45	1.2384	5.45	1.6956	7.45	2.0018	9.45	2.2460
1.50	0.4054	3.50	1.2527	5.50	1.7047	7.50	2.0149	9.50	2.2513
1.55	0.4382	3.55	1.2669	5.55	1.7138	7.55	2.0215	9.55	2.2565
1.60	0.4700	3.60	1.2809	5.60	1.7228	7.60	2.0281	9.60	2.2618
1.65	0.5007	3.65	1.2947	5.65	1.7316	7.65	2.0347	9.65	2.2670
1.70	0.5306	3.70	1.3083	5.70	1.7405	7.70	2.0412	9.70	2.2721
1.75	0.5596	3.75	1.3217	5.75	1.7491	7.75	2.0477	9.75	2.2773
1.80	0.5877	3.80	1.3350	5.80	1.7578	7.80	2.0541	9.80	2.2824
1.85	0.6151	3.85	1.3480	5.85	1.7664	7.85	2.0605	9.85	2.2875
1.90	0.6418	3.90	1.3610	5.90	1.7750	7.90	2.0668	9.90	2.2925
1.95	0.6678	3.95	1.3737	5.95	1.7834	7.95	2.0731	9.95	2.2976
2.00	0.6931	4.00	1.3863	6.00	1.7918	8.00	2.0794	10.00	2.3026
2.05	0.7178	4.05	1 3987	6.05	1.8000	8.05	2.0857	10.25	2.3273
2.10	0.7419	4.10	1.4010	6.10	1.8083	8.10	2.0918	10.50	2.3514
2.15	0.7654	4.15	1.4231	6.15	1.8164	8.15	2.0988	10.75	2.3749
2.20	0.7885	4.20	1.4351	6.20	1.8245	8.20	2.1041	11.00	2.3979
2.25	0.8110	4.25	1.4469	6.25	1.8326	8.25	2.1102	12.00	2.4849
2.30	0.8329	4.30	1.4586	6.30	1.8405	8.30	2.1162	13.00	2.5626
2.35	0.8544	4.35	1.4701	6.35	1.8484	8.35	2.1222	14.00	2.6390
2.40	0.8755	4.40	1.4816	6.40	1.8563	8.40	2.1282	15.00	2.7103
2.45	0.8961	4.45	1.4929	6.45	1.8640	8.45	2.1342	16.00	2.7751
2.50	0.9163	4.50	1.5040	6.50	1.8718	8.50	2.1400	17.00	2.8332
2.55	0.9361	4.55	1.5151	6.55	1.8795	8.55	2.1459	18.00	2.8903
2.60	0.9555	4.60	1.5260	6.60	1.8870	8.60	2.1518	19.00	2.9444
2.65	0.9746	4.65	1.5369	6.65	1.8946	8.65	2.1576	20.00	2.9957
2.70	0.9932	4.70	1.5475	6.70	1.9021	8.70	2.1633	21.00	3.0445
2.75	1.0116	4.75	1.5581	6.75	1.9095	8.75	2.1690	22.00	3.0910
2.80	1.0296	4.80	1.5686	6.80	1.9169	8.80	2.1747	23 00	3.0355
2.85	1.0473	4.85	1.5790	6.85	1.9242	8.85	2.1804	24.00	3.1780
2.90	1.0647	4.90	1.5892	6.90	1.9315	8.90	2.1860	25.00	3.2189
2.95	1.0818	4.95	1.5994	6.95	1.9387	8.95	2.1916	30.00	3.3782

clearance space, expressed in cubic feet, will give the volume of steam in the cylinder, which multiplied by the weight per cubic foot of steam at the pressure as measured will give the weight of steam consumed during one stroke. From this should be deducted the steam saved by compression as shown by the diagram, in order to get a true measure of the economy of the engine. Having thus determined the weight of steam consumed for one stroke, multiply it by twice the number of strokes per minute and by 60, which will give the total weight consumed per hour. This divided by the horse-power will give the rate per horse-power per hour.

Cylinder Condensation and Reëvaporation. When the exhaust valve opens to permit the exit of the steam, there is a perceptible cooling of the walls of the cylinder, especially in condensing engines when a high vacuum is maintained. This results in more or less condensation of the live steam admitted by the opening of the steam valve; but if the exhaust valve is caused to close at the proper time so as to retain a portion of the steam to be compressed by the piston on the return stroke, a considerable portion of the water caused by condensation will be reëvaporated into steam by the heat and consequent rise in pressure caused by compression.

Ordinates. Parallel lines drawn at equal distances apart across the face of the diagram, and perpendicular to the atmospheric line. They serve as a guide to facilitate the measurement of the average forward pressure throughout the stroke, or the pressure at any point of the stroke if desired.

Eccentric. A mechanical device used in place of a crank for converting rotary into reciprocating motion. An eccentric is in fact a form of crank in which the crank

pin, corresponding to the eccentric sheave, embraces the shaft, but owing to the great leverage at which the friction between the sheave and the strap acts, compared with its short turning leverages, it can only be used to advantage for the purpose named above.

Eccentric Throw is the distance from the center of the eccentric to the center of the shaft. This definition also applies to the term "radius of eccentricity."

Eccentric Position. The location of the highest point of the eccentric relative to the center of the crank pin, measured or expressed in degrees.

Angular Advance. The distance that the high point of the eccentric is set ahead of a line at right angles with the crank. In other words, the lap angle, plus the lead angle. If a valve had neither lap nor lead, the position of the high point of the eccentric would be on a line at right angles with the crank; as, for instance, the crank being at 0° the eccentric would stand at 90°.

Valve Travel. The distance covered by the valve in its movement. It equals twice the throw of the eccentric.

Lap. The amount that the ends of the valve project over the edges of the ports when the valve is at mid-travel.

Outside or Steam Lap. The amount that the end of the valve overlaps or projects over the outside edge of the steam port.

Inside Lap. The lap of the inside, or exhaust edge of the valve over the inside edge of the port.

Lead. The amount that the port is open when the crank is on the dead center. The object of giving a valve lead is to supply a cushion of live steam which, in conjunction with that already confined in the clearance space by compression, shall serve to bring the moving parts of

the engine to rest quietly at the end of the stroke, and also quicken the action of the piston in beginning the return stroke.

Compression. Closing of the exhaust passage before the steam is entirely exhausted from the cylinder. A certain quantity of steam is thus compressed into the clearance space.

Table 52, giving areas and circumferences of circles, is here inserted, for the reason that in the study of indicator diagrams there is very often occasion for reference to such a table.

In the following analysis of indicator diagrams all of the illustrations are reproductions of actual diagrams taken under ordinary working conditions.

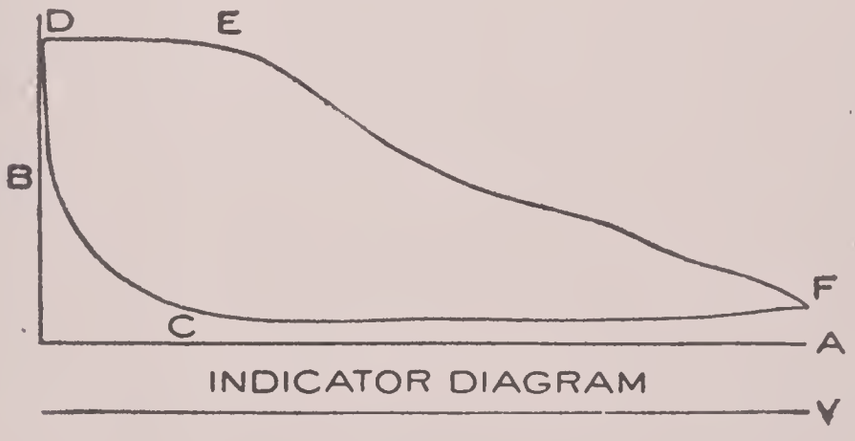


Fig. 115.

Fig. 115 shows a sample diagram taken from a locomotive running at a speed of 29 miles per hour, and cutting off at 5½ inches. By reference to the letters, the different lines and points into which an indicator diagram is divided may be readily distinguished. The line V indicates the base line, or line of perfect vacuum, from which pressures are measured, especially in calculations of steam consumed per horse-power hour. This

line is drawn at a point 14.7 lbs. below the atmospheric line A, as indicated by measurements made with the scale adapted to the spring used in taking the diagram. The method of drawing the line of atmospheric pressure has already been described. It is from this line that the mean effective pressure of the steam upon the piston during the stroke is estimated in all calculations of diagrams taken from locomotives and other non-condensing engines.

Admission at the beginning of the stroke is shown at B, and from B to D is the admission line. From D to E is the steam line. E is the point of cut-off, and from E to F is the expansion curve. F is the point of release or exhaust opening, and from this point to C is the line of back pressure or counter pressure, and the amount of this pressure depends upon the height of this line above line A. Compression begins at C, and from this point to B is the compression curve.

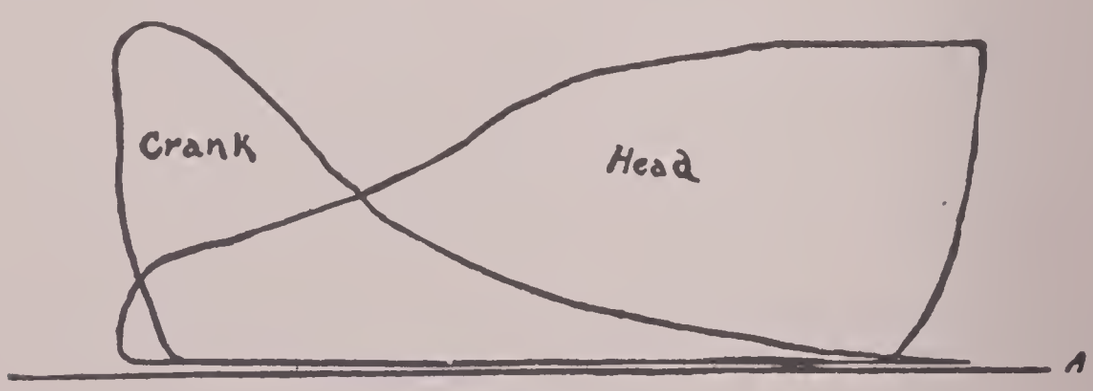


Fig. 116.

Fig. 116 shows bad valve adjustment, the engine doing by far the largest portion of the work in the head end of the cylinder.

In order to illustrate the process of ascertaining the M. E. P. without dividing the diagram into ordinates, the following computation is given, together with rules, etc. In this process two important factors are necessary, viz.,

Table 52.—Areas and Circumferences of Circles.

Diam.	Area.	Circum.	Diam.	Area.	Circum.
.25	.049	.7854	15.5	188.692	48.694
.5	.1963	1.5708	16	201 062	50.265
1.0	.7854	3.1416	16.25	207 394	51.051
1.25	1.2271	3.9270	16.5	213.825	51.836
1.5	1.7671	4.7124	17	226 980	53.407
2	3.1416	6.2832	17.25	233.705	54.192
2.25	3.9760	7.0686	17.5	240.520	54.978
2.5	4.9087	7.8540	18	254.469	56.548
3	7.0686	9.4248	18.25	261.587	57.334
3.25	8.2957	10.210	18.5	268.803	58.119
3.5	9.6211	10.995	19	283 529	59.690
4	12.566	12.566	19.25	291.039	60.475
4.25	14.186	13.351	19.5	298.648	61.261
4.5	15.904	14.137	20	314.160	62.832
5	19.635	15.708	20.25	322.063	63.617
5.25	21.647	16.493	20.5	330.064	64.402
5.5	23.758	17.278	21	346.361	65.973
6	28.274	18.849	21.25	354.657	66.759
6.25	30.679	19.635	21.5	363.051	67.544
6.5	33.183	20.420	22	380.133	69.115
7	38.484	21.991	22.25	388.822	69.900
7.25	41.282	22.776	22.5	397.608	70.686
7.5	44.178	23.562	23	415.476	72.256
8	50.265	25.132	23.25	424.557	73.042
8.25	53.456	25.918	23.5	433.731	73.827
8.5	56.745	26.703	24	452.390	75.398
9	63.617	28.274	24.25	461.864	76.183
9.25	67.200	29.059	24.5	471.436	76.969
9.5	70.882	29.845	25	490.875	78.540
10	78.540	31.416	25.25	500.741	79.325
10.25	82.516	32.201	25.5	510.706	80.110
10.5	86.590	32.986	26	530.930	81.681
11	95.033	34.557	26.25	541.189	82.467
11.25	99.402	35.343	26.5	551.547	83.252
11.5	103.869	36.128	27	572.556	84.823
12	113.097	37.699	27.25	583.208	85.608
12.25	117.859	38.484	27.5	593.958	86.394
12.5	122.718	39.270	28	615.753	87.964
13	132.732	40.840	28.25	626.798	88.750
13.25	137.886	41.626	28.5	637.941	89.535
13.5	143.130	42.411	29	660.521	91.106
14	153.938	43.982	29.25	671.958	91.891
14.25	159.485	44.767	29.5	683.494	92.677
14.5	165.130	45.553	30	706.860	94.248
15	176.715	47.124	30.25	718.690	95.033
15.25	182.654	47.909	30.5	730.618	95.818

Table 52.—*Continued.*

Diam.	Area.	Circum.	Diam.	Area.	Circum.
31	754.769	97.389	46.5	1698.23	146 08
31.25	766.992	98.175	47	1734.94	147.65
31.5	799.313	98.968	47.25	1753.45	148.44
32	804.249	100.53	47.5	1772.05	149.22
32.25	816.86	101.31	48	1809.56	150.79
33	855.30	103.67	48.25	1828.46	151.58
33.25	868.30	104.45	48.5	1847.45	152.36
33.5	881.41	105.24	49	1885.74	153.93
34	907.92	106.81	49.25	1905.03	154.72
34.25	921.32	107.60	49.5	1924.42	155.50
34.5	934.82	108.38	50	1963.50	157.08
35	962.11	106.95	50.25	1983.18	157.86
35.25	975.90	110.74	50.5	2002.96	158 65
35.5	989 80	111.52	51	2042.82	160.22
36	1017.8	113.09	51.25	2062.90	161.00
36.25	1032.06	113.88	51.5	2083.07	161.79
36.5	1046.35	114.66	52	2123.72	163.36
37	1075.21	116.23	52.25	2144.19	164.14
37.25	1089.79	117.01	52.5	2164.75	164.19
37.5	1104.46	117.81	53	2206.18	166.50
38	1134.11	119.38	53.25	2227.05	167.29
38.25	1149.08	120.16	53.5	2248.01	168.07
38.5	1164.15	120.95	54	2290.22	169.64
39	1194.59	122.52	54.25	2311.48	170.43
39.25	1209.95	123.30	54.5	2332.83	171.21
39.5	1225.42	124.09	55	2375.83	172.78
40	1256.64	125.66	55.25	2397.48	173.57
40.25	1272.39	126.44	55 5	2419 22	174.35
40.5	1288.25	127.23	56	2463.01	175.92
41	1320.25	128.80	56.25	2485.05	176.71
41.25	1336.40	129.59	56.5	2507.19	177.5
41.5	1352.65	130 37	57	2551.76	179.07
42	1385.44	131.94	57.25	2574.19	179.85
42.25	1401.98	132.73	57.5	2596.72	180.64
42.5	1418.62	133.51	58	2642.08	182.21
43	1452.20	135.08	58.25	2664.91	182.99
43.25	1469.13	135 87	58.5	2687.83	183.78
43.5	1486.17	136.65	59	2733 97	185.35
44	1520.53	138.23	59.25	2757.19	186.14
44.25	1537.86	139.01	59.5	2780.51	186.92
44.5	1555.25	139.80	60	2827.44	188.49
45	1590.43	141.37	60.25	2851.05	189.28
45.25	1608.15	142.15	60.5	2874.76	190.06
45.5	1625.97	142.94	61	2922.47	191.64
46	1661.90	144.51	61.25	2946.47	192.42
46.25	1680.01	145.29	61.5	2970.57	193.21
			62	3019.07	194.78

Table 52.—*Continued.*

Diam.	Area.	Circum.	Diam.	Area.	Circum.
62.25	3043.47	195.56	78	4778.37	245.04
62.5	3067.96	196.35	78.25	4809.05	245.83
63	3117.25	197.92	78.5	4839.83	246.61
63.25	3142.04	198.71	79	4901.68	248.19
63.5	3166.92	199.50	79.25	4932.75	248.97
64	3216.99	201.06	79.5	4963.92	249.76
64.25	3242.17	201.85	80	5026.56	251.33
64.5	3267.46	202.68	80.5	5089.58	252.90
65	3318.31	204.20	81	5153.00	254.47
65.25	3343.88	204.99	81.5	5216.82	256.04
65.5	3369.56	205.77	82	5281.02	257.61
66	3421.20	207.34	82.5	5345.62	259.18
66.25	3447.16	208.13	83	5410.62	260.75
66.5	3473.23	208.91	83.5	5476.00	262.32
67	3525.66	210.49	84	5541.78	263.89
67.25	3552.01	211.27	84.5	5607.95	265.46
67.5	3578.47	212.06	85	5674.51	267.04
68	3631.68	213.63	85.5	5741.47	268.60
68.25	3658.44	214.41	86	5808.81	270.17
68.5	3685.29	215 20	86.5	5876.55	271.75
69	3739 28	216.77	87	5944.66	273.32
69.25	3766.43	217.55	87.5	6013.21	274.89
69.5	3793.67	218.34	88	6082.13	276.46
70	3848.46	219.91	88.5	6151.44	278.03
70.25	3875.99	220.70	89	6221.15	279.60
70.5	3903.63	221.48	89.5	6291.25	281.17
71	3959.20	223.05	90	6371.64	282.74
71.25	3987 13	223.84	90.5	6432.62	284.31
71.5	4015.16	224.62	91	6503.89	285.88
72	4071 51	226.19	91.5	6573.56	287.46
72.25	4099.83	226.98	92	6647.62	289.03
72.5	4128.25	227.75	92.5	6720.07	290.60
73	4185.39	229.34	93	6792.92	292.17
73.25	4214.11	230.12	93.5	6866.16	293.74
73.5	4242.92	230.91	94	6939.79	295.31
74	4300.85	232.48	94.5	7013.81	296.88
74.25	4329.95	233.26	95	7088.23	298.45
74.5	4359.16	234.05	95.5	7163.04	300.02
75	4417.87	235.62	96	7238.25	301.59
75.25	4447.37	236.40	96.5	7313.80	303.16
75.5	4476.97	237.19	97	7389.81	304.73
76	4536.37	238.76	97.5	7466.22	306.30
76.25	4566.36	239.55	98	7542.89	307.88
76.5	4596.35	240.33	98.5	7620.09	309.44
77	4656.63	241.90	99	7697.70	311.02
77.25	4686 92	242.69	99.5	7775.63	312.58
77.5	4717.30	243.47	100	7854.00	314.16

the absolute initial pressure and the absolute terminal pressure, and they can both be obtained from the diagram by measuring with the scale adapted to the spring used. Thus, in Fig. 117 the absolute initial pressure measured from the line of perfect vacuum V to line B is 77 lbs., and the absolute terminal pressure measured from V to line B′ is 21 lbs. The ratio, or number of expansions, is found thus:

Rule. Divide the absolute initial pressure by the absolute terminal pressure; thus, $77 \div 21 = 3.65 =$ number of expansions.

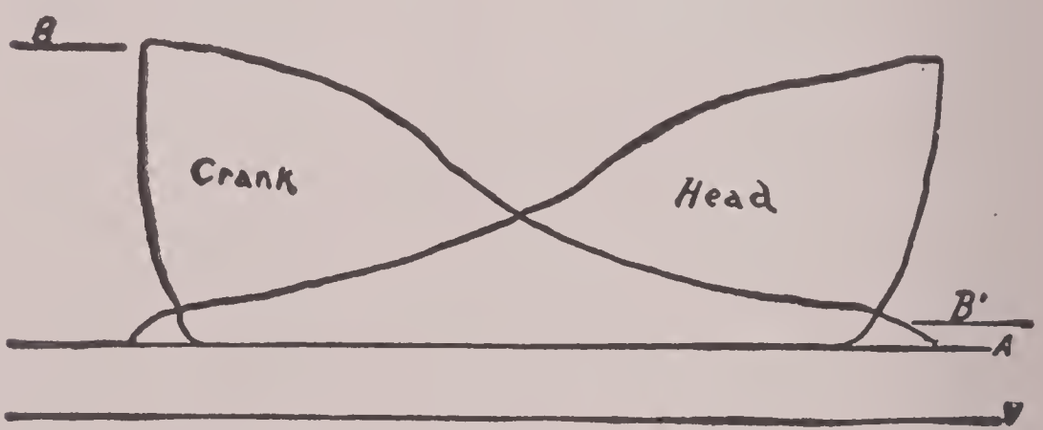


Fig. 117.

Second. Find mean forward pressure.

Rule. Multiply absolute initial pressure by the hyperbolic logarithm of number of expansions plus 1, and divide product by number of expansions. Thus, referring to Table 51, it will be seen that the hyperbolic logarithm of 3.65 is 1.2947, to which 1 must be added. Then $\frac{77 \times 2.2947}{3.65} = 48.4$ lbs., which is the absolute mean forward pressure. From this deduct the absolute back pressure, which is 16 lbs. or 1 lb. above atmosphere; thus, $48.4 - 16 = 32.4$ lbs. M. E. P.

Third. Find I. H. P.

Area of piston minus one-half area of rod $\times$ M. E. P. $\times$ piston speed in feet per minute, divided by 33,000. Thus (the diameter of rod being 3 in.),

$$\frac{250.96 \times 32.4 \times 564}{33{,}000} = 138.9 \text{ I. H. P.}$$

The steam consumption per I. H. P. per hour may also be computed by means of Table 53, which was originally calculated by Mr. Thomson, and is based upon the following theory:

Table 53.

T. P.	W.	T. P.	W.	T. P.	W
3	117.30	13	466.57	23	798.10
3.5	135.75	13.5	483.43	23.5	814.39
4	153.88	14	500.22	24	830.64
4.5	171.94	14.5	517.07	24.5	846.96
5	186.75	15	533.85	25	863.25
5.5	207.60	15.5	550.64	25.5	879.49
6	225.24	16	567.36	26	895.70
6.5	242.97	16.5	584.10	26.5	911.86
7	260.54	17	600.78	27	927.99
7.5	278.06	17.5	617.40	27.5	944.07
8	295.44	18	633.96	28	960.12
8.5	312.80	18.5	650.46	28.5	976.27
9	330.03	19	666.90	29	992.38
9.5	347.27	19.5	683.38	29.5	1008.46
10	364.40	20	699.80	30	1024.50
10.5	381.57	20.5	716.27	30.5	1040.51
11	398.64	21	732.69	31	1056.48
11.5	415.73	21.5	749.06	31.5	1072.42
12	432.72	22	765.38	32	1088.32
12.5	449.69	22.5	781.76	32.5	1104.35

A horse-power = 33,000 ft. lbs. per minute, or 1,980,000 ft. lbs. per hour, or 1,980,000 × 12 = 23,760,000 in. lbs. per hour, meaning that the same amount of energy required to lift 33,000 lbs. one foot high in one minute of time would lift 23,760,000 lbs. one inch high in one minute of time. Now, if an engine were driven by a fluid that weighed one pound per cubic inch, and the mean effective pressure of this fluid upon the piston was one pound per square inch, it would require 23,760,000 lbs. of the fluid per horse-power per hour. But, if in place of the heavier fluid we substitute pure distilled water, of which it requires 27.648 cu. in. to weigh one pound, the consumption per I. H. P. per hour will be considerably less; as, for instance, 23,760,000 ÷ 27.648 = 859,375 lbs., which would be the rate per hour of the water driven engine if the M. E. P. of the water was one pound per square inch and if the M. E. P. was increased to 20 lbs., then twenty times more power would be developed with the same volume of water, but the weight of water consumed per H. P. per hour would be proportionately less. Now, if the engine is driven by steam it will consume just as much less water in proportion as the water required to make the steam is less in volume than the steam used. Therefore if the above constant number, 859,375, be divided by the M. E. P. of any diagram and by the volume of the terminal pressure, the quotient will be the water (or steam) consumption per I. H. P. per hour.

Referring to Table 53, the numbers in the W columns are the quotients obtained by dividing the constant, 859.375, by the volumes of the absolute pressures given in the columns under T. P. and which represent terminal pressures. The table is considerably abridged from the original, which was very full and complete, the pressures ad-

vancing by tenths of a pound from 3 lbs. to 60 lbs.; but it is seldom that in ordinary practice there is needed such accuracy. If at any time, however, a diagram should show a terminal pressure not given in the table, the correct factor for that pressure can be easily found by dividing the constant 859,375 by the relative volume of the pressure as found in Table 4 of the properties of saturated steam given in another Section.

Referring again to Fig. 117, it is seen that the terminal pressure is 21 lbs. absolute, and by reference to Table 53 and glancing down column T. P. until 21 is reached, it will be seen that the number opposite in column W is 732.69. This number divided by the M. E. P. of the diagram Fig. 117, which is 32.4 lbs., gives 22.6 lbs. per I. H. P. per hour as the steam consumption. The rate thus found makes no allowance for clearance and compression, however, and these two very important items will be treated in a succeeding section, together with the method of correction for the above, viz., clearance and compression, as they enter largely into the steam economy of an engine.

Steam Consumption from Indicator Diagrams. In calculating the steam consumption of an engine, two very important factors must not be lost sight of, viz., clearance and compression. Especially is this the case in regard to clearance when there is little or no compression, for the reason that the steam required to fill the clearance space at each stroke of the engine is practically wasted, and all of it passes into the atmosphere or the condenser, as the case may be, without having done any useful work except to merely fill the space devoted to clearance. On the other hand, if the exhaust valve is closed before the piston completes the return stroke, the steam then re-

maining in the cylinder will be compressed into the clearance space and can be deducted from the total volume, which, without compression, would have been exhausted at the terminal pressure.

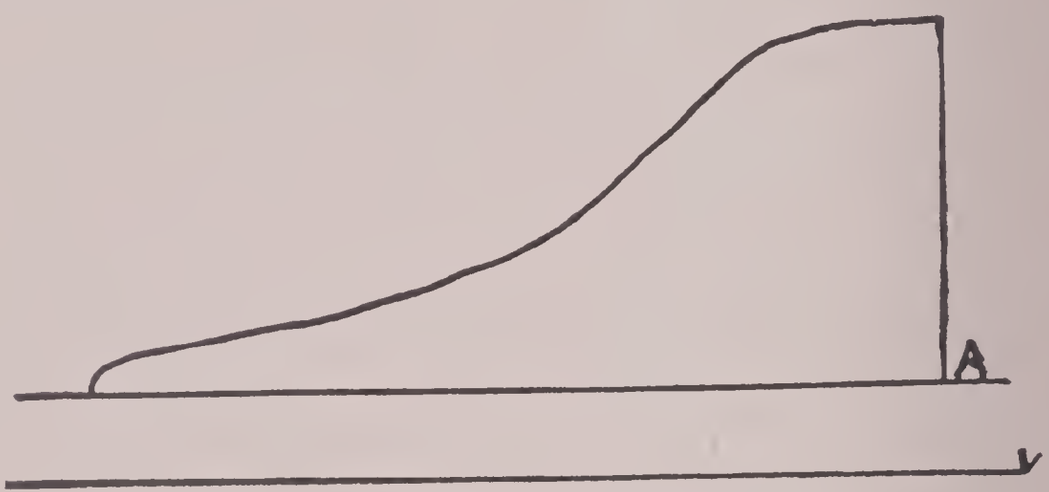


Fig. 118.

Figs. 118 and 119, which are reproductions of diagrams taken by the author while adjusting the valves on a 16 × 42 in. corliss engine, will serve to graphically illustrate this point. Fig. 118, which was the first one to be

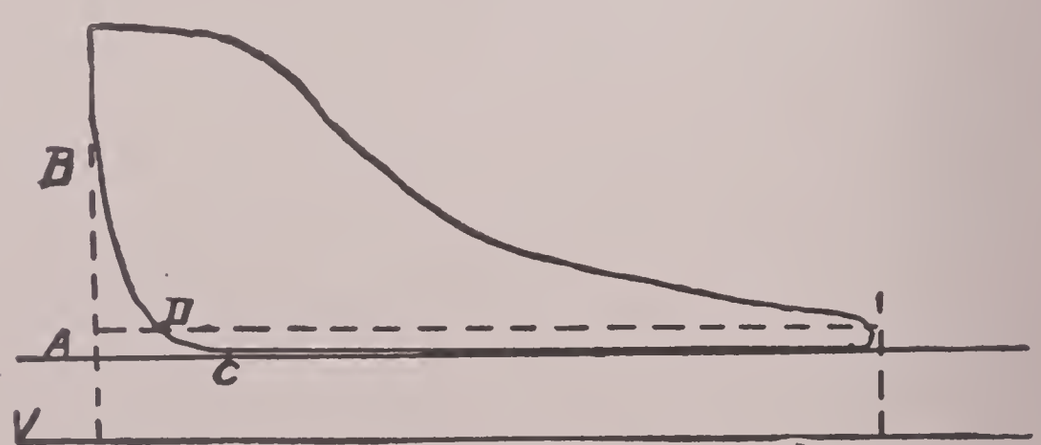


Fig. 119.

taken, shows no compression. The point of admission at A is plainly defined by the square corner at the extreme end of the stroke. The clearance of this engine

is 4 per cent of the volume of the piston displacement. The engine being 16 in. bore, by 42 in. stroke, the piston displacement is found by the following calculation: Area of piston, 201.06 sq. in. × stroke, 42 in. = 8444.52 cu. in. The volume of clearance space is equal to 8444.52 cu. in. × .04 = 337.78 cu. in., which divided by 1,728 = .195 cu. ft.

By reference to Fig. 119, taken after adjusting the valves for compression, it will be noticed that the steam is there compressed to 37 lbs., the compression curve beginning at C and ending at B. There is therefore compressed during each stroke a volume of steam equal to .195 cu. ft. at a pressure of 37 lbs. gauge, or 52 lbs. absolute.

One cubic foot of steam at 52 lbs. absolute pressure weighs .1243 lbs., and .195 cu. ft. will weigh .1243 × .195 = .0242 lbs.

The engine was running at 70 R. P. M., or 140 strokes per minute. Thus, according to Fig. 119, the total weight of steam compressed and doing useful work during one hour, and which without compression would have passed out through the exhaust pipe, is equal to .0242 × 140 × 60 = 203.28 lbs.

Now, in order to estimate the steam consumption of the above engine from diagram Fig. 118, it would be necessary to account for all the steam occupying not only the volume of the piston displacement at the end of the stroke, but the clearance as well, for the reason, as before stated, that it would all be released before exhaust closure. This would equal 8444.52 cu. in. + 337.78 cu. in. = 8782.3 cu. in., which divided by 1,728 = 5.08 cu. ft. each stroke, or 10.16 cu. ft. each revolution.

The absolute terminal pressure of Fig. 118 is 20 lbs.

One cubic foot of steam at this pressure weighs .0507 lbs., and the weight of steam consumed each revolution would therefore be $10.16 \times .0507 = .515$ lbs., which multiplied by 70 R. P. M. = 36.05 lbs. per minute, or 2,163 lbs. per hour. The horse-power developed by the engine at the time was 80. Therefore the steam consumption per I. H. P. per hour $= 2{,}163 \div 80 = 27$ lbs.

Referring again to Fig. 119, it will be remembered that the total weight of steam compressed during one hour was 203.28 lbs. The weight of steam consumed per hour, therefore, equals $2{,}163 - 203.28 = 1959.7$ lbs.

Owing to compression, the work area of Fig. 119 is somewhat smaller than that of Fig. 118, amounting in fact to the area of the irregular figure enclosed between the points A, B and C. The work represented by this figure amounts to a very small proportion of the total work indicated by Fig. 118, still, in order to arrive at correct conclusions, it should be deducted therefrom.

Assuming the negative work to be equal to .55 horse-power, we have $80 - .55 = 79.45$ I. H. P. as the work represented by Fig. 119. As the total weight of steam consumed in one hour was 1959.7 lbs., the steam consumption per I. H. P. per hour will be $1959.7 \div 79.45 = 24.67$ lbs., a saving by compression of 2.33 lbs. per H. P. per hour, besides the great advantage of having a cushion of steam in contact with the piston at the termination of the stroke, thus bringing the moving parts of the engine to rest quietly without shock or jar.

The steam consumption may also be computed from the diagram, regardless of the dimensions of the cylinder or the horse power developed. The mean effective pressure and also the absolute terminal pressure must, however, be known. This method has been referred to,

but in the computation therein made, no correction was made for clearance and compression.

Having reviewed these two factors at considerable length, it will now be in order to more fully explain the methods of treating diagrams when it is desired to make these corrections.

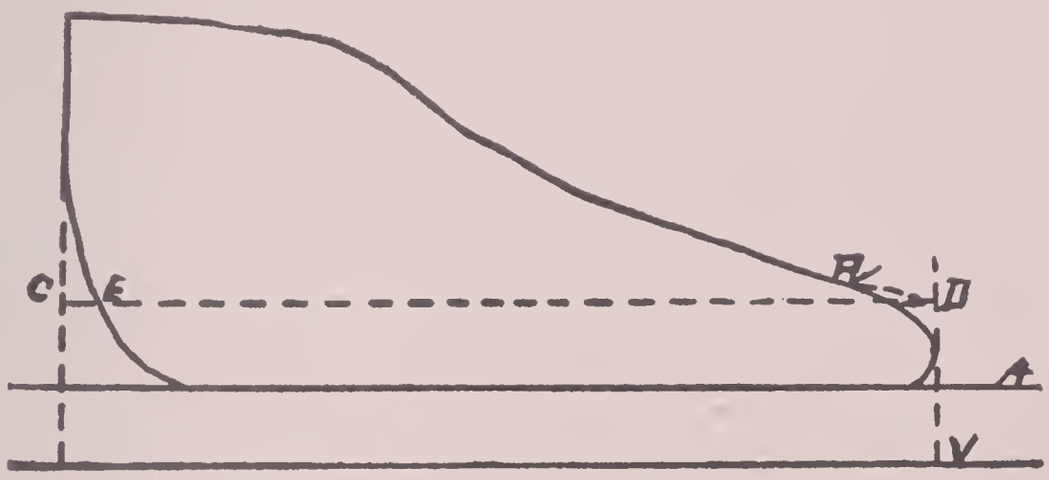


Fig. 120.

First, draw vertical lines C and D, Fig. 120, at each end of the diagram, and perpendicular to the atmospheric line. Draw line V, representing perfect vacuum, 14.7 lbs. below the atmospheric line, as indicated on the scale adapted to the diagram, which in this case is 50 lbs. to the inch. Continue the expansion from R, where release begins, until it intersects line D V, from which point the absolute terminal pressure can be measured.

Having ascertained the terminal pressure, which for Fig. 120 is 30 lbs., draw line D E, which may be called the consumption line for 30 lbs. The terminal being 30 lbs., refer to Table 53 and find in column W, opposite 30 in column T. P., the number 1,024.5. Divide this number by the M. E. P., which in Fig. 120 is 41 lbs., and the quotient, which is 24.99 lbs., is the uncorrected

rate of steam consumption. This rate stands for the total consumption throughout the whole stroke represented on the diagram by the distance from D to C, which measures 3.25 in., but it is evident that there is a small portion of the return stroke, that indicated by the distance from E to C, during which the steam compressed in the clearance space should not be charged to the consumption rate, but should be deducted therefrom. In order to do this, multiply the uncorrected rate by the distance from D to E, which is 3⅛ in., or 3.125 in., and divide the product by the distance from D to C, 3¼ in., or 3.25 in. Thus, $24.99 \times 3.125 \div 3.25 = 24.03$ lbs., which is the corrected rate and represents a saving by compression of $24.99 - 24.03 = .96$ lbs., or nearly 3.7 per cent.

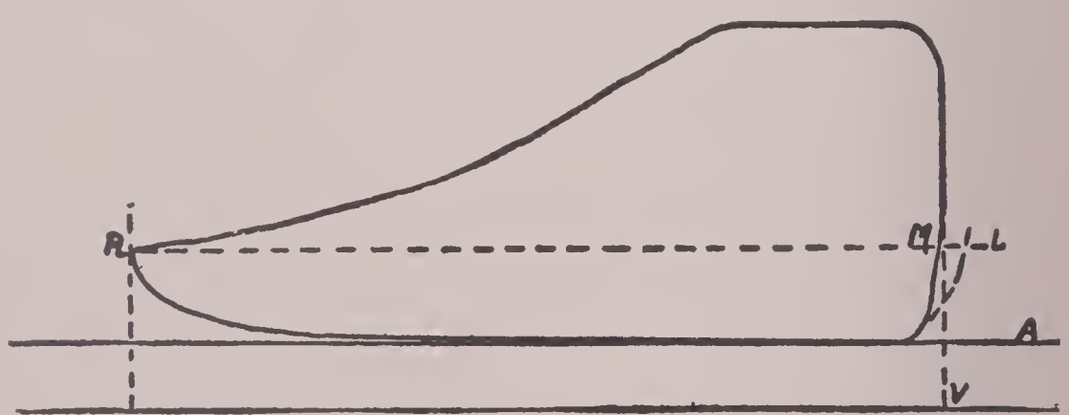


Fig. 121.

In many cases the terminal pressure greatly exceeds the compression, an illustration of which is given in Fig. 121. It now becomes necessary to extend the compression curve to L, a point equidistant from the vacuum line with the terminal at R. The consumption line R. L. now becomes longer than the stroke line R. M.; therefore the corrected rate will exceed the uncorrected rate by just so much; as, for instance, terminal pressure = 34 lbs. The factor, as per Table 53, = 1152.26, and the

M. E. P. of the diagram is 47 lbs. Then, 1,152.26 ÷ 47 = 24.5 lbs., uncorrected rate; 24.5 × 3.125 in. (distance R. L.) ÷ 3 in. (distance R. M.) = 25.52 lbs., corrected rate, a loss of a little more than one pound, or about 4 per cent.

There is another class of diagrams very frequently encountered, in which the terminal pressure is considerably below the compression curve, and in order to compute the consumption rate by the above method it becomes necessary to continue the compression curve downwards until it meets the terminal, as illustrated at A, Fig.

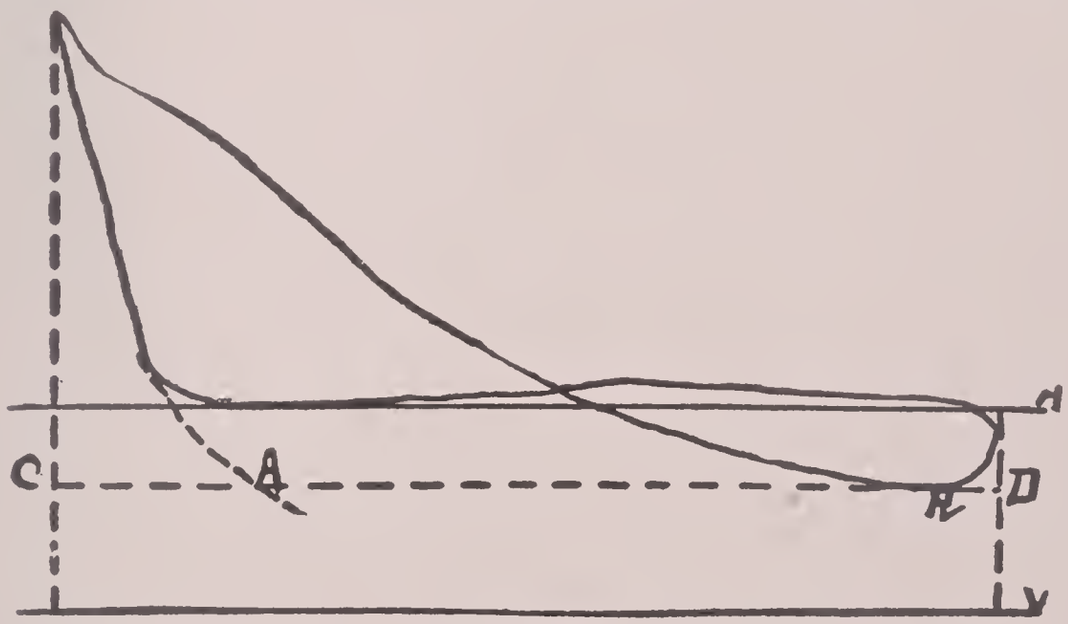


Fig. 122.

122. R is the point of release, D A represents the consumption line, and D C the stroke. The terminal is 8.5 lbs., and the factor for that pressure, according to Table 53, is 312.8. Dividing this number by the M. E. P., which was 7 lbs., gives 44.6 lbs. as the uncorrected rate. The distance D to A, where the compression curve intersects the consumption line, is 2.625 in., and the total length of the diagram C to D is 3.375 in. Then 44.6 × 2.625 ÷ 3.375 = 35 lbs. as the corrected rate.

Theoretical Clearance. The expansion and compression curves of a diagram are created by the expansion and compression of all the steam admitted during the stroke. This includes the steam in the clearance space as well as in the cylinder proper. It is evident, therefore, that the volume of the clearance is one of the factors controlling the form of these curves, and when the clearance is known a correct expansion or isothermal curve may be theoretically constructed, as will be explained later on. Also, if the actual curves, either expansion or compression, of a diagram assume an approximately correct form, the clearance, if not already known, may be determined theoretically from them; although too much confidence should not be put in the results, as they are liable to show either too little, or too much clearance, generally the latter, especially if figured from the compression curve.

For the benefit of those who may desire to test this method of ascertaining the percentage of clearance of their engines, several illustrations will be given of its application to actual diagrams taken from engines in which the clearance was known.

Fig. 123 is from an engine in which the clearance was known to be 5 per cent. As compression cuts but a very small figure in this diagram, the expansion curve alone will be utilized for obtaining the theoretical clearance, and the process is as follows:

Select two points, C and R, in the curve as far apart as possible, but be sure that they are each within the limits of the true curve. Thus C is located just after cut-off takes place, and R is at a point just before release begins. From C draw line C D parallel with the atmospheric line. From D draw line D R, and from C draw line C E, both perpendicular to the atmospheric

line. Then from R draw line R E, forming a rectangular parallelogram, C D R E, with two opposite corners, C and R, within the curve. Now through the other two corners, D and E, draw the diagonal D E, extending it downwards until it intersects the vacuum line V. From this point erect the vertical line V W, which is the theoretical clearance line.

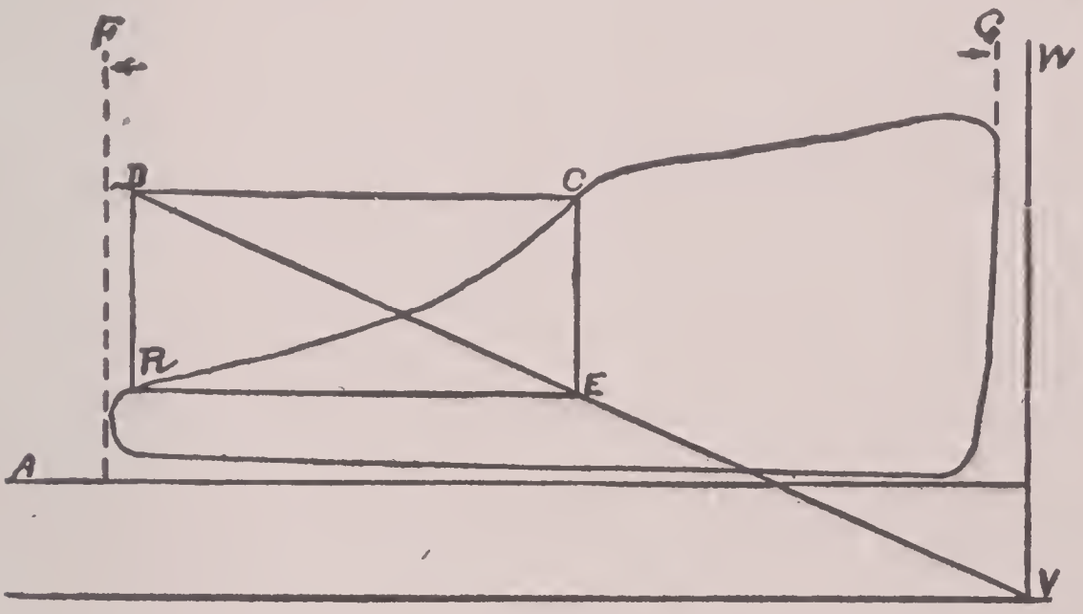


Fig. 123.

To prove the result, proceed as follows: Measure the length of diagram from F to G, which in this case is 3.75 in., representing piston displacement. Next measure the distance from F to the clearance line V W, which is 3.91 in., representing piston displacement with volume of clearance added. Then $3.91 - 3.75 = .16$, which represents volume of clearance; and $.16 \times 100 \div 3.75 =$ 4.3 per cent, which is approximately near the actual clearance, which, as before stated, was 5 per cent.

The Theoretical Expansion Curve. According to Boyle's law the volume of all elastic gases is inversely as their pressures, and steam, being a gas, conforms

substantially to this law; although the expansion curves of indicator diagrams are affected more or less by the loss of heat transmitted through the cylinder walls, and by the change in the temperature of the steam produced by the changes in pressure during the progress of the stroke. The pressure generally falls more rapidly during the first part of the stroke, and less rapidly during the last portion than it should, in order to conform strictly to the above law, and the terminal pressure usually is greater than it should be to agree with the ratio of expansion. But this fullness of the expansion curve of the diagram near the end compensates in a measure for the too rapid fall near the beginning of the stroke. Therefore, if the engine is in fairly good condition, with the valves properly adjusted and not leaking, and the piston rings are steam-tight, it may be assumed that the expansion of the steam in the cylinder takes place according to Boyle's law, and it is found that the expansion curve drawn by the indicator practically coincides with a hyperbolic curve constructed according to that law.

Fig. 124 graphically illustrates the application of the hyperbolic law to the expansion of gases. The horizontal lines represent volumes and the vertical lines represent pressures. The base line, A F, represents the full stroke of a piston in the cylinder of an engine, and the vertical line A I represents the pressure of the steam at the commencement of the stroke.

Suppose there is no clearance and that the steam has been admitted up to point H when it is cut off. The rectangle A B H I is the product of the pressure multiplied by the volume of the steam thus admitted. When the piston has traveled from A to C the volume of the steam has been doubled and the pressure C L has been

reduced to just one-half what it was at A I, but the area of the rectangle A C L M is equal to the area of the initial rectangle, and, as before, is the product of the pressure C L multiplied by the volume A C. As the piston travels still farther, as from A to D, the steam is expanded to four volumes, while the pressure at D K will only be one-fourth that of the initial pressure; but the new rectangle A D K N is still equal in area to either of the others, A B H I or A C L M.

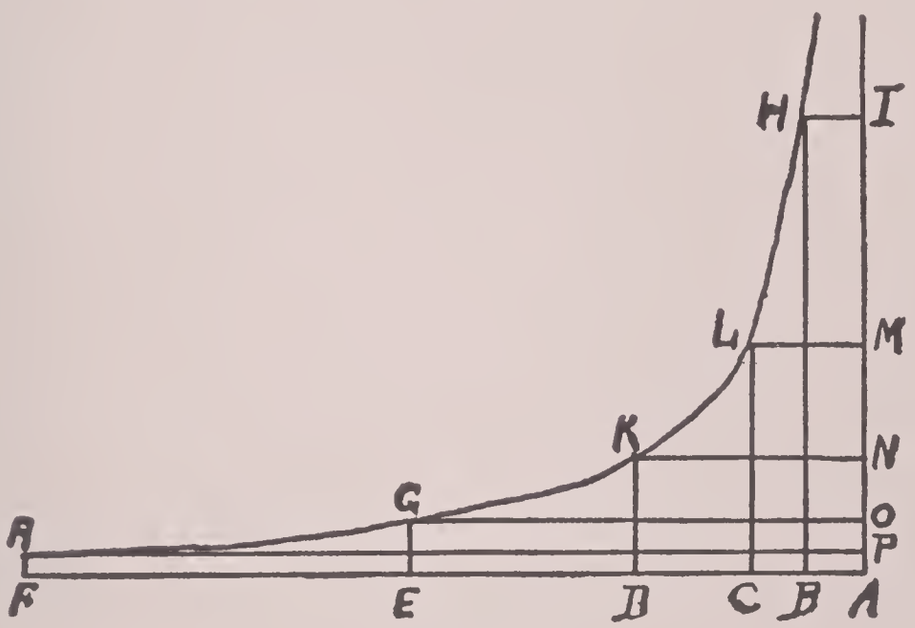


Fig. 124.

The same law applies to each of the remaining rectangles; A E G O representing five volumes and one-fifth of the initial pressure, and A F R P representing six times the initial volume and one-sixth of the initial pressure, but each having the same area as the initial rectangle A B H I. Now, the area of the rectangle A B H I represents the work done by the steam up to the point of cut-off, and the area of the hyperbolic figure enclosed by the lines B H R F represents the work done by the expansion of the steam after cut-off occurs. This

area and the amount of work it represents may be computed by means of the known relations of hyperbolic surfaces with their base lines; as, for instance, if the base lines A B, A C, A D, etc., extend in geometrical ratio. as 1, 2, 4, 8, 16, etc., the successive areas, B H L C, B H K D, B H G E, etc., increase in arithmetical ratio, as 1, 2, 3, 4, etc.

On the principles of common logarithms, which represent in arithmetical ratio, natural numbers in geometrical ratio, tables of hyperbolic logarithms have been computed for the purpose of facilitating the calculation of areas of work due to different degrees of expansion. Such a table is given elsewhere in this book, and the method of calculating the M. E. P. by this means is described.

ISOTHERMAL CURVE.

A theoretical curve may be constructed conjointly with the actual expansion curve of a diagram by first locating the clearance and vacuum lines and then pursuing the method illustrated by Fig. 125. A curve so produced is called an isothermal curve, meaning a curve of the same temperature.

Referring to Fig. 125, suppose, first, that it is desired to ascertain how near the expansion curve of the diagram coincides with the isothermal curve, at or near the point of cut off. Select point R near where release begins, but still well within the expansion curve. From this point draw the vertical line, R T, parallel with the clearance line, V S. Then draw the horizontal line, S T. parallel with the atmospheric line, and at such a height above it as will equal the boiler pressure as measured by the scale adapted to the diagram; such measurement to be made from the atmospheric line to correspond with the

gauge pressure. From T draw the diagonal T V, and from R draw the horizontal line R D parallel with the atmospheric line. From D, where this line intersects T V, erect the perpendicular D E, thus forming the parallelogram R D E T, and as line T V passes through two of its opposite angles and meets the junction of the clearance and vacuum lines, the other two angles, R and E, will be in the theoretical curve, and R being the starting point, it is obvious that this curve must pass through E, which would be the theoretical point of cut-off on the steam line S T.

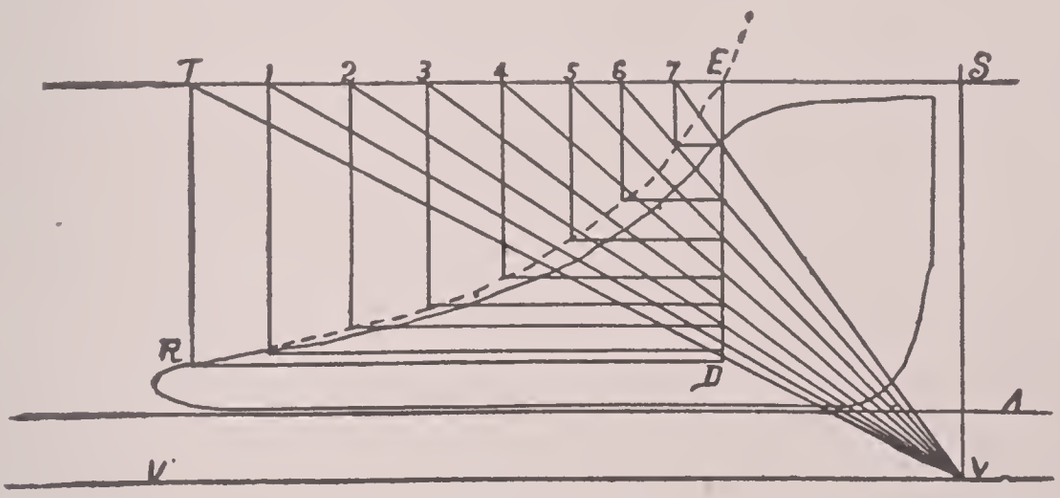


Fig. 125.

Two important points in the theoretical curve have now been located, viz., E as the cut-off, and R as the point of release. In order to obtain intermediate points, draw any desired number of lines downward from points in S T, as 1, 2, 3, 4, 5, etc., and continue them downwards far enough to be sure that they will meet the intended curve, and from the same points in S T draw diagonals 1 V, 2 V, 3 V, 4 V, 5 V, etc., all to converge accurately at V. From the intersection of these diagonals with D E draw horizontal lines parallel with V V', and the points of junction of these lines with the vertical

lines will be points in the theoretical curve. It will now be an easy matter to trace the curve through these points. If, on the other hand, it be desired to compare the curves toward the exhaust end of the diagram, draw lines E D and E T, Fig. 126, also T R, locating R near where release commences, after which draw line R D, completing the parallelogram E T R D, fixing R as a point in the theoretical curve started at E. After drawing the diagonal T V, proceed in the same manner as before to locate the intermediate points.

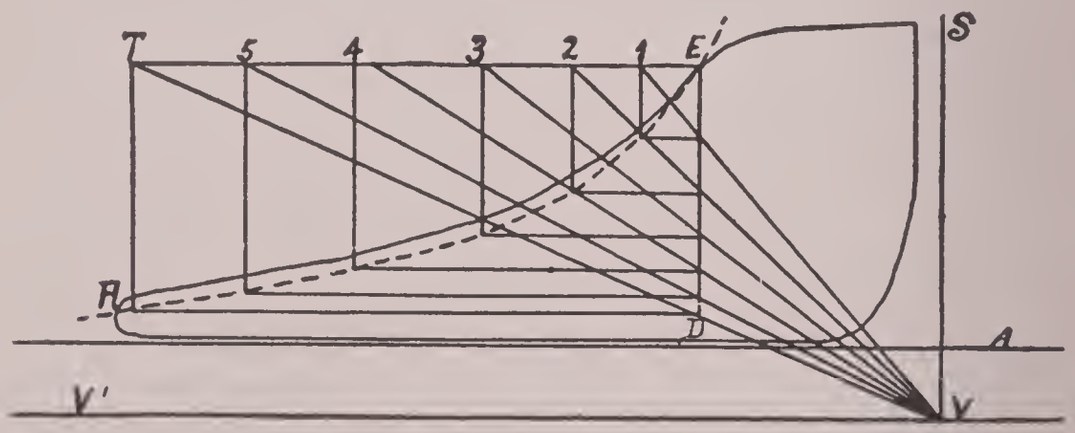


Fig. 126.

It will be observed that in order to ascertain the performance of the steam near the beginning of the stroke, the starting point of the isothermal curve must be near the point of release, and conversely, if the starting point of the curve is located near the point of cut-off and coincident with the actual curve, the test will apply towards the end of the stroke. It is not to be expected that the expansion curve of any diagram taken in practice will conform strictly to the lines of the isothermal curve, especially towards the latter end of the stroke, owing to the reëvaporation of water resulting from the condensation of steam which was retained in the cylinder by the

closing of the exhaust valve. This reëvaporation commences just as soon as the temperature of the steam owing to reduction of pressure due to expansion, falls below the temperature of the cylinder walls, and it continues at an increasing rate until release occurs. The tendency of this reëvaporation or generation of steam within the cylinder during the latter portion of the stroke is to raise the terminal pressure considerably above what it would be if true isothermal expansion took place. The terminal pressure may also be augmented by a leaky steam valve, while, on the other hand, a leaky piston would cause a lowering of the terminal and an increase in the back pressure.

The Adiabatic Curve. If it were possible to so protect or insulate the cylinder of a steam engine that there would be absolutely no transmission of heat either to or from the steam during expansion, a true adiabatic curve or "curve of no transmission" might be obtained. The closer the actual expansion curve of a diagram conforms to such a curve, the higher will be the efficiency of the engine as a machine for converting heat into work.

Fig. 127 illustrates a method of figuring a curve which, while not strictly adiabatic, will be near enough for all practical purposes, while at the same time it will give the student an opportunity to study the laws governing the expansion of saturated steam.

To draw the curve, first locate the clearance and vacuum lines V S and V V'. Next locate point R in the expansion curve near where release begins, making this the starting point, and also the point of coincidence of the expansion curve with the adiabatic curve.

The other points in the curve are located from the volumes of steam at different pressures during expan-

sion; the pressures being measured from the line of perfect vacuum, and the volumes from the clearance line.

The absolute pressure at R, Fig. 127, is 26 lbs. From point R erect the perpendicular R T. Also draw horizontal line R 26 parallel with the vacuum line and at a height equal to 26 lbs. above vacuum line V V', as shown by the scale, which in this case was 40. The length of line R 26, measured from R to the clearance line, is 3 1-16 in., or 3.0625 in. By reference to Table 4 it will be seen that the volume of steam at 26 lbs. absolute, as compared with

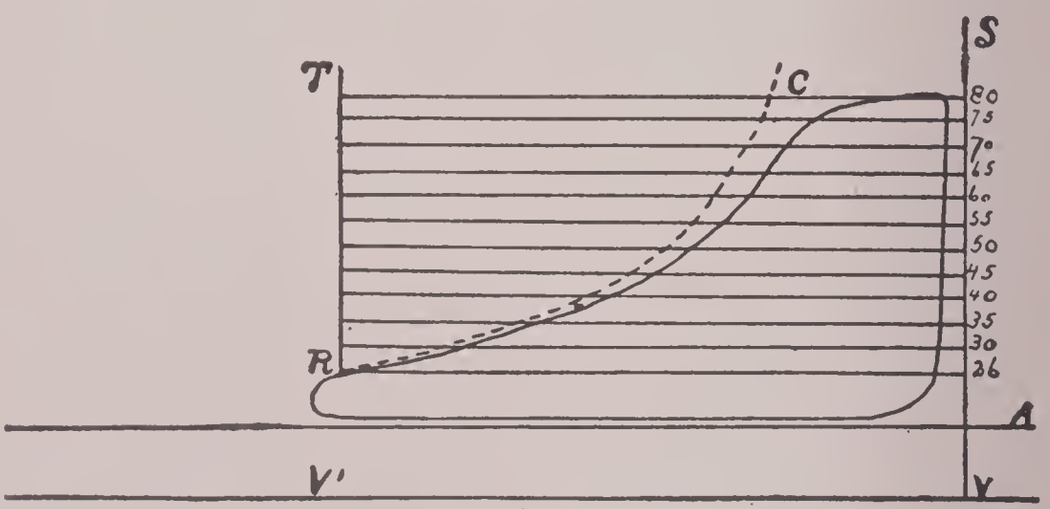


Fig. 127.

water at 39°, is 962. Now, if the length of line R 26 be divided by this volume, and the quotient multiplied by each of the volumes of the other pressures represented at points 30, 35, 40, 45, etc., up to the initial pressure, the products will be the respective distances from the clearance line of points in the adiabatic curve. These points can be marked on the horizontal lines drawn from the clearance line, to line R T.

Starting with line R 26, it has been noted that its length is 3.0625 in., and that the volume was 962. $3.0625 \div 962 = .003$. Then the volume of steam at 30 lbs. is 841,

which being multiplied by .003 = 2.5 in., the length of line 30. Next, the volume at 35 lbs. = 728. Multiplying this volume by .003 = 2.1 in., length of line 35, and so in like manner for each of the other points.

The process involves considerable figuring and careful and accurate measurements, which should be made with a steel rule with decimal graduations. It is not expected that the cut Fig. 127 will be found accurate enough in its measurements to serve as a standard; it being intended only to serve as an illustration of the process. The diagram from which the illustration was drawn was taken from a 600 H. P. engine situated some 200 ft. from the boilers, and there was a considerable cooling of the steam by the time it reached the engine, the effect of which is apparent. The curve produced by the measurements is shown by the broken line. The process can be applied to any diagram.

Power Calculations. The area of the piston (minus one-half the area of rod) multiplied by the M. E. P., as shown by the diagram, and this product multiplied by the number of feet traveled by the piston per minute (piston speed), will give the number of foot-pounds of work done by the engine each minute, and if this product be divided by 33,000, the quotient will be the indicated horse-power (I. H. P.) developed by the engine.

Therefore one of the first requisites in power calculations is to ascertain the M. E. P. Beginning with the most simple, though only approximately correct, method of obtaining the average pressure, as illustrated by Fig. 128, draw line A B touching at A and cutting the diagram in such manner that the space D above it will equal in area spaces C and E taken together, as nearly as can be estimated by the eye. Then with the scale measure the

pressure along the line F G at the middle of the diagram, which will be the M. E. P.

The process is based upon the theory that the average width of any tapering figure is its width at the middle of its length. This method should not be relied upon as accurate, but is convenient at times when it is desired to make a rough estimate of the horse-power of an engine.

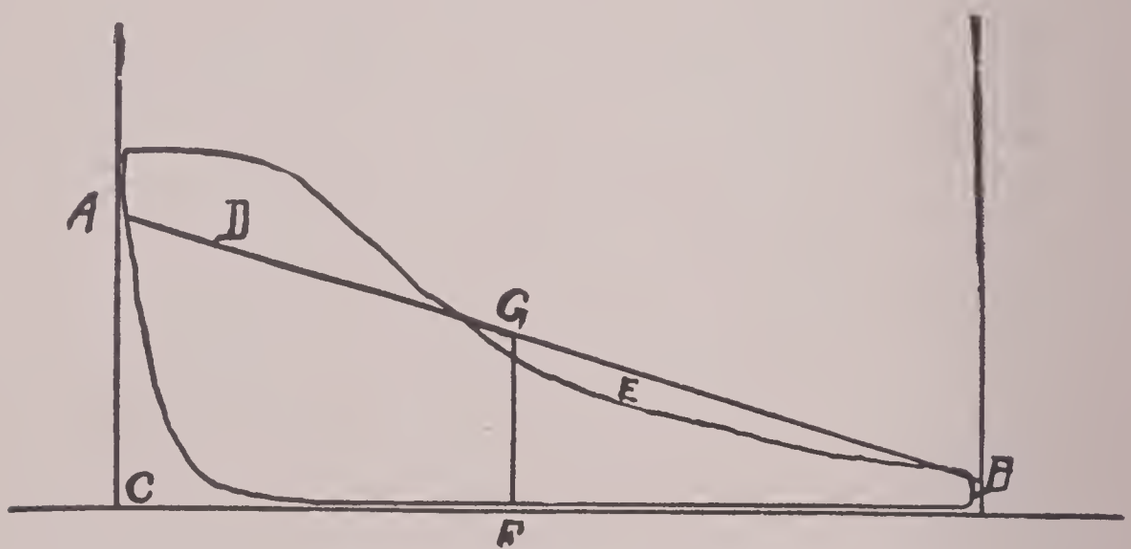


Fig. 128.

Figuring the M. E. P. by Ordinates. This is a very common method and one which can be relied upon to give accurate results, provided care is exercised in its use.

The process consists in drawing any convenient number of vertical lines perpendicular to the atmospheric line across the face of the diagram, spacing them equally, with the exception of the two end spaces, which should be one-half the width of the others, for the reason that the ordinates stand for the centers of equal spaces, as, for instance, line 1, Fig. 129, stands for that portion of the diagram from the end to the middle of the space between it and line 2. Again, line 2 stands for the remain-

ing half of the second space and the first half of the third, and so on. This is an important matter, and should be thoroughly understood, because if the spaces are all made of equal width, and measurements are taken on the ordinates, the results will be incorrect, especially in the case of high initial pressure and early cut-off, following which the steam undergoes great changes.

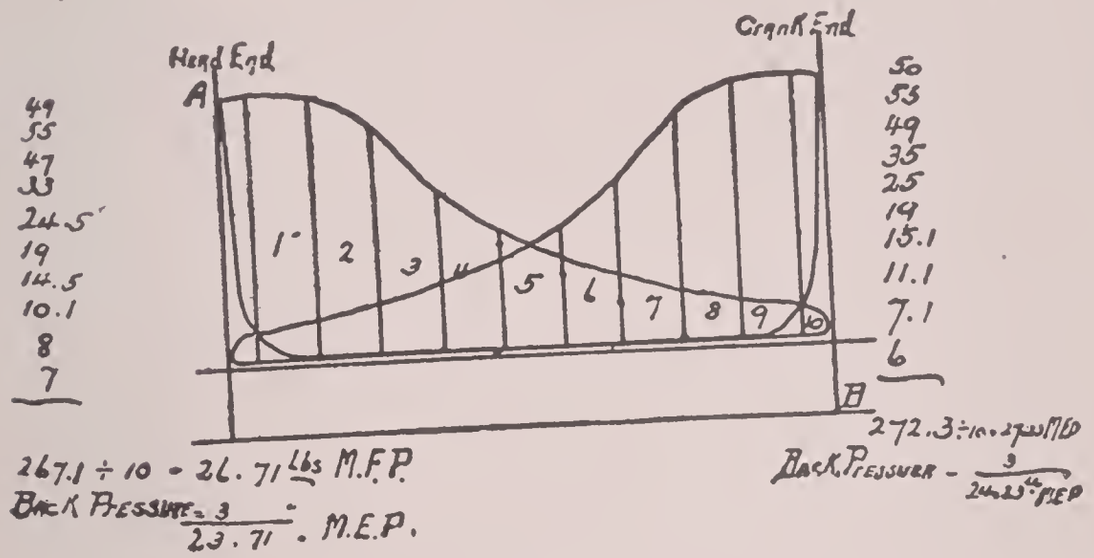


Fig. 129.

If the spaces are all made equal, the measurements will require to be taken in the middle of them, and errors are liable to occur, whereas, if spaced as before described, the measurements can be made on the ordinates, which is much more convenient and will insure correct results. Any number of ordinates can be drawn, but ten is the most convenient and is amply sufficient, except in case the diagram is excessively long. For spacing the ordinates, dividers may be used, or a parallel ruler may be procured from the makers of the indicator; but one of the most convenient and easily procurable instruments for this purpose is a common two-foot rule, and the method of using it is illustrated in Fig. 129.

First draw vertical lines at each end of the diagram, perpendicular to the atmospheric line and extending downwards to the vacuum line, or below it, if necessary, in order to have a point on which to lay the rule. In Fig. 129, points A and B are found to be the most convenient. Now lay the rule diagonally across the diagram, touching at A and B, and the distance will be found to be 3¾ in., or 60 sixteenths.

Suppose it be desired to draw 10 ordinates. Divide 60 by 10, which will give 6 sixteenths, or ⅜ in., as the width of the spaces, but as the two end spaces are to be one-half the width of the others, there will be 11 spaces altogether, the two outer ones having a width equal to one-half of ⅜, or 3-16. Now apply the rule again in the same manner, touching at points A and B, and with a sharp pointed pencil begin at A and mark the location of the first ordinate according to the rule, at a distance of 3-16th from the end. Then ⅜ from this mark make another one, which will locate the second ordinate, and proceed in like manner to locate the others. The last two or three marks generally come below the diagram, and if the diagram be taken from a condensing engine it may be necessary to tack it on to a larger sheet of paper in order to get these points. Having correctly located the ordinates, they may now be drawn perpendicular to the atmospheric line or vacuum line, either of which will answer.

It should be noted that, owing to the diagonal position of the rule with relation to the atmospheric line, the spaces are not of the actual width as described by the rule, but this is unimportant, so long as they are of a uniform width. This method can be applied to any diagram, no matter what its length may be, and point B may be lo-

cated at any distance below the atmospheric or vacuum lines, wherever it is the most convenient for the subdivisions on the rule, sixteenths, eighths, etc., so long as it is in line with the end of the diagram. Having thus drawn the ordinates, the M. E. P. may be found by measuring the pressure expressed by each one, using for this purpose the scale adapted to the spring used, adding all together and dividing by the number of ordinates which will give the average pressure.

Referring to Fig. 129, begin with ordinate No. 1 on the diagram, from the head end of the cylinder. In this case a 40 spring was used. Lay the scale on the ordinate with the zero mark where it intersects the compression curve. The pressure is seen to be 49 lbs. Set this down at that end of the card and measure the pressure along ordinate No. 2, which is 55 lbs. Proceed in this manner to measure all the ordinates, placing the resulting figures in a column, after which add them together and divide by 10. The result is 26.71 lbs., which is the mean forward pressure (M. F. P.). To obtain the mean effective pressure, deduct the back pressure, which is represented by the distance of the exhaust line of the diagram above the atmospheric line in a non-condensing engine, and in a condensing engine the back pressure is measured from the line of perfect vacuum, 14.7 lbs., according to the scale below the atmospheric line.

In Fig. 129 the back pressure is found to be 3 lbs. Therefore the M. E. P. of the head end will be 26.71 — 3 = 23.71 lbs. On the crank end the M. F. P. is 27.23 lbs., and 27.23 — 3 = 24.23 lbs. = M. E. P. The average effective pressure on the piston, therefore, will be 23.71 + 24.23 ÷ 2 = 23.97 lbs.

Unless great care is exercised in the measurements,

errors are liable to occur in applying this method, especially with scales representing high pressures, as 60, 80, etc. The most convenient and reliable method is to take a narrow strip of paper of sufficient length, and starting at one end, apply its edge to each ordinate in succession and mark their lengths on it consecutively with the point of a knife blade or a sharp pencil. Having thus marked on the paper the total length of all the ordinates, ascertain the number of inches and fractions of an inch thereon, the fractions to be expressed decimally, and divide by the number of ordinates. The quotient will be the average height of the diagram, and as the scale expresses the number of pounds pressure for each inch or fraction of an inch in height, if the average height of the diagram be multiplied by the number of the scale, the product will be the M. F. P.

Referring again to Fig. 129, if the lengths of the ordinates drawn on the head end diagram be measured, their sum will be found to be 6 8-12th or 6.666 in. Dividing this by 10 gives .666 in. as the average height. The mean forward pressure will then be as follows: .666 $\times$ 40 = 26.64 lbs., or practically the same as found by the other method.

Fig. 130 illustrates a type of diagram frequently met with, and one which requires somewhat different treatment in estimating the power developed. It will be noticed that, owing to light load and early cut-off, the expansion curve drops considerably below the atmospheric line, notwithstanding that the engine from which this diagram was taken is a non-condensing engine. When release occurs at R, and the exhaust side of the piston is exposed to the atmosphere, the pressure immediately rises to a point equal to, or slightly above, that of the atmosphere.

Fig. 130 was taken during a series of experiments made by the author for the purpose of ascertaining the friction of shafting and machinery, and the engine it was obtained from is a Buckeye 24 x48 in. The boiler pressure at the time was only 40 lbs., and a No. 20 spring was used. The ordinates are drawn according to the method illustrated in Fig. 129. By placing the rule on points A and B, the distance between those two points is found to be 3⅝ in., or 58 sixteenths. Dividing this by 10 gives 5.8 sixteenths, or nearly ⅜ in., as the width of the spaces; the two end spaces being one-half of this, or 3-16th in. wide. The first five ordinates, counting from A, express forward pressure, represented by the arrows. The re-

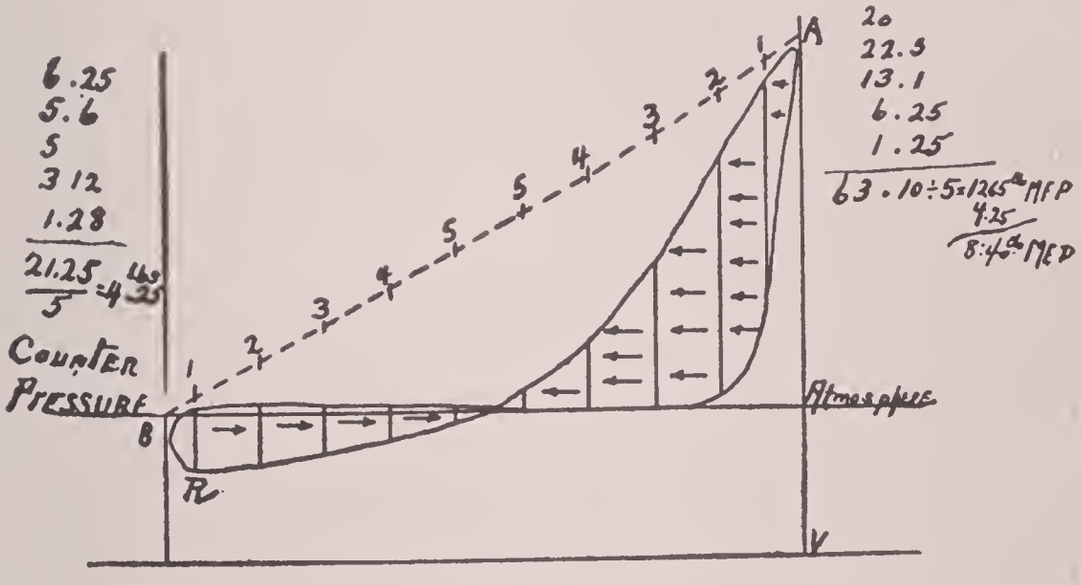


Fig. 130.

maining five ordinates, counting from B, express counter or back pressure, represented by the arrows pointing in the opposite direction. Measuring the pressures along the first five ordinates, and adding them together, gives 63.1 lbs., which divided by 5 gives 12.65 lbs. as the mean forward pressure (M. F. P.).

Then figuring up the counter pressure in the same manner on the other five ordinates, beginning at B, the result is 4.25 lbs. The M. E. P., therefore, will be 12.65 — 4.25 =8.4 lbs.

Obtaining the M. E. P. with the Planimeter. The area of the diagram represents the actual work done by the steam acting upon the piston. In a non-condensing engine the lower or exhaust line of the diagram must be either coincident with or slightly above the atmospheric line in order to express positive work. Any deviation of this line, either above or below the atmospheric line, represents counter pressure, the amount of which may be ascertained by measurements with the scale, and should be deducted from the mean forward pressure.

On the other hand, the exhaust line of a diagram from a condensing engine falls more or less below the atmospheric line, according to the degree of vacuum maintained, and the nearer this line approaches the line of perfect vacuum, as drawn by the scale, 14.7 lbs. below the atmospheric line, the less will be the counter pressure, which in this case is expressed by the distance the exhaust line is above that of perfect vacuum.

The prime requisite, therefore, in making power calculations from indicator diagrams is to obtain the average height or width of the diagram, supposing it were reduced to a plain parallelogram instead of the irregular figure which it is.

The planimeter, Fig. 131, is an instrument which will accurately measure the area of any plane surface, no matter how irregular the outline or boundary line is, and it is particularly adapted for measuring the areas of indicator diagrams, and in cases where there are many diagrams to work up, it is a very convenient instrument and

saves much time and mental effort. In fact, the planimeter has of late years become an almost indispensable adjunct of the indicator. It shows at once the area of the diagram in square inches and decimal fractions of a square inch, and when the area is thus known it is an easy matter to obtain the average height by simply dividing the

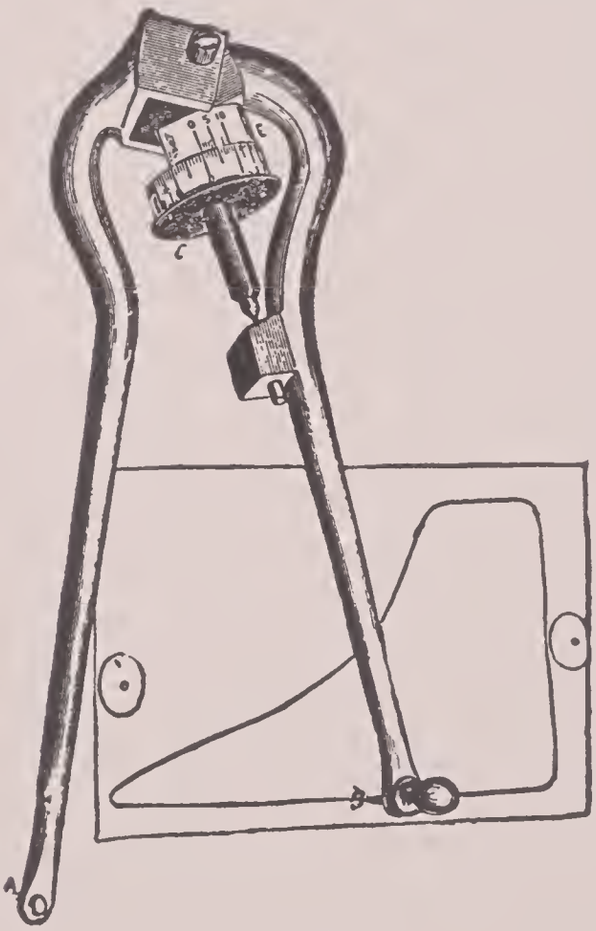


Fig. 131.

area in inches by the length of the diagram in inches. Having ascertained the average height of the diagram in inches or fractions of an inch, the mean or average pressure is found by multiplying the height by the scale. Or the process may be made still more simple by first multiplying the area, as shown by the planimeter in square

inches and decimals of an inch, by the scale, and dividing the product by the length of the diagram in inches. The result will be the same as before, and troublesome fractions will be avoided.

Questions.

364. Who invented the indicator, and for what purpose did he apply it to his engine?

365. What are the principles governing the action of the indicator?

366. What will a truthful diagram from a steam engine cylinder show?

367. Describe in general terms the construction of an indicator.

368. Does the steam act upon both sides of the indicator piston?

369. What does the atmospheric line show?

370. Is this line important in the study of indicator diagrams?

371. Where should the line of back pressure appear in a diagram from a non-condensing engine?

372. What controls the length of stroke of the indicator piston?

373. What does the number on the spring mean?

374. Should the pencil fall below the atmospheric line in a diagram from a locomotive?

375. What is the most practical device for reducing the motion of the crosshead to correspond with the motion of the indicator drum?

376. What are the main requirements in indicator connections?

377. What should be done with these pipes before attaching the indicator?

378. What regulates the height of the diagram?

379. What is a convenient rule to be observed in the selection of the spring?

380. What governs the length of the diagram?

381. Describe the best method of tracing the atmospheric line.

382. What data should be noted on the diagram as soon as taken?

383. By what means may the taking of indicator diagrams from locomotives be greatly facilitated?

384. What is absolute pressure?

385. What is gauge pressure?

386. What is initial pressure?

387. What is terminal pressure and how may it be ascertained theoretically?

388. What is back pressure?

389. What is absolute back pressure?

390. What is meant by ratio of expansion?

391. What does the term wire-drawing mean when applied to an indicator diagram?

392. What is condenser pressure?

393. What does the term vacuum imply?

394. What is absolute zero?

395. What is meant by the term piston displacement?

396. What is piston clearance?

397. What is steam clearance?

398. What is a horse-power?

399. What is meant by piston speed?

400. Define Boyle's law of expanding gases.

401. What is an adiabatic curve?

402. What is an isothermal curve?

403. What is the first law of thermodynamics?

404. What is the unit of work?

405. Define the first law of motion?

406. What is momentum?

407. What is the maximum theoretical duty of steam?

408. What is meant by the term steam efficiency?

409. How may the term engine efficiency be defined?

410. What are common logarithms?

411. What are hyperbolic logarithms, and how are they found?

412. What are ordinates as applied to an indicator diagram?

413. What is an eccentric?

414. What is meant by the throw of an eccentric?

415. What is meant by position of the eccentric?

416. What is angular advance?

417. What is meant by the expression, steam consumption of an engine?

418. What effect has back pressure upon the work of an engine?

419. What relation should the steam line of a diagram bear to the atmospheric line ?

420. In calculations for steam consumption what two important factors must be considered?

421. How is the piston displacement of an engine ascertained?

422. What do the expansion and compression curves of a diagram show?

423. Is steam a gas?

424. What effect does reëvaporation have upon the expansion curve?

425. How is the horse-power of an engine calculated?

426. What is meant by the expression, M. E. P.?

427. What is a planimeter?

COMPOUND LOCOMOTIVES.

The principal object in compounding locomotives is to effect economy in fuel, and this economy is due to the fact that with the compound engine the steam may be expanded to a much lower pressure than is possible with the simple engine, before it is allowed to exhaust into the atmosphere. Another source of economy in compounding the cylinders of any steam engine, stationary or locomotive, is the prevention of that excessive condensation which is sure to result when steam at a high pressure is admitted to a cylinder, the walls of which are at a comparatively low temperature at the moment of admission, and this takes place at each stroke of the simple engine; as, for instance, assume the initial pressure to be 195 lbs., and the pressure at release to be 8 lbs. The temperature of steam at 195 lbs. is 385°, and at 8 lbs. pressure the temperature is 235°. This drop of 150° in the temperature during each stroke of the engine, tends to cool the walls of the cylinder, which will be warmed again by the next admission of steam. A large amount of heat is thus being continually absorbed by the cylinder walls, and there is also a constant loss caused by condensation.

In the compound locomotive the expansion of the steam is divided between two cylinders, proportioned in such a way that the amount of work done in each will be the same.

Various types of compound locomotives have been designed and built by eminent engineers in this country and in Europe, and while, as before stated, the main

object in compounding is to utilize as much of the tremendous energy stored in the coal as it is possible to utilize, still there are other important problems to be solved in the design and operation of compound locomotives, not the least of which is to so proportion the cylinders, especially of a cross compound, that there will be an equal distribution of power on each side of the engine, or, in other words, that the engine will be balanced. Another problem that has been constantly before the purchaser and the builder of compound locomotives, is that of keeping the number of parts down to as low a figure as possible, and thus to produce a machine that will use steam on the compound principle, and yet at the same time eliminate as far as possible the liability of additional expense for repairs that has always been connected with the compound, as compared with the simple engine.

The progress along these lines has been slow, but there has been a marked development in the right direction, and there is no doubt that compound locomotives have come to stay, and that eventually they will become the standard type. It therefore behooves engine men (engineers and firemen) to study them, and endeavor to familiarize themselves with their construction and operation.

There are in use at the present time, in this country, four separate and distinct types of compound locomotives, each having its peculiar features. First, there is the Vauclain compound. This is a four cylinder engine, having two cylinders on each side of the engine. One of these cylinders is a high-pressure and the other a low-pressure cylinder, one being located directly above the other.

Second, the balanced compound, a four cylinder en-

gine, the two high-pressure cylinders being located under the center of the smoke arch, between the frames, and the two low-pressure cylinders on the outside.

Third, the tandem compound, a four cylinder engine, having one high, and one low-pressure cylinder on each side, these cylinders being in line with each other, and served by one piston rod, thus bringing all the strains in direct line also.

Fourth, the cross compound, a two cylinder engine, having the high-pressure cylinder on one side, and the low-pressure cylinder on the opposite side, the diameter or bore of the cylinders being proportioned in such manner that an equal amount of power will be developed on both sides. This ratio is generally one to three, that is, the area of the low-pressure piston is about three times that of the high, for the reason that the initial pressure of the steam admitted to the low-pressure cylinder is greatly reduced below the point at which it entered the high-pressure cylinder, and requires a larger area of piston to act upon in order to produce the same amount of power that it did in the high-pressure cylinder. These four forms of compound locomotives will be taken up, and each discussed in its regular order. The same valve gear is used upon compound locomotives as upon simple engines, or those in which there is but single expansion.

VAUCLAIN COMPOUND LOCOMOTIVE.

The Vauclain compound locomotive is the invention of Mr. Samuel M. Vauclain of the Baldwin Locomotive Works, and the following description of this system of compounding has been mainly furnished by the Baldwin Locomotive Works of Philadelphia, Pa.

In designing the Vauclain system of compound locomotives, the aim has been:

1. To produce a compound locomotive of the greatest efficiency, with the utmost simplicity of parts and the least possible deviation from existing practice. To realize the maximum economy of fuel and water.

2. To develop the same amount of power on each side of the locomotive, and avoid the racking of machinery resulting from unequal distribution of power.

3. To insure at least as great efficiency in every respect as in a single-expansion locomotive of similar weight and type.

4. To insure the least possible difference in cost of repairs.

5. To insure the least possible departure from the method of handling single-expansion locomotives; to apply equally to passenger or freight locomotives for all gauges of track, and to withstand the rough usage incidental to ordinary railroad service.

The principal features of construction are as follows:

Cylinders. The cylinders consist of one high-pressure and one low-pressure for each side, the ratio of the volumes being as nearly three to one as the employment of convenient measurements will allow. They are cast in one piece with the valve-chamber and saddle, the cylinders being in the same vertical plane, and as close together as they can be with adequate walls between them.

Where the front rails of the frames are single bars, the high-pressure cylinder is usually put on top, as shown in Fig. 132, but when the front rails of frames are double, the low-pressure cylinder is usually on top, as shown in Fig. 133.

The former (Fig. 132) is used in "eight-wheel" or

American type passenger locomotives, and in "ten-wheel" locomotives, while the latter (Fig. 133) is used in Mogul, Consolidation and Decapod locomotives; for the various other classes of locomotives the most suitable arrangement is determined by the style of frames.

Fig. 132.

Fig. 133.

Fig. 134 shows the arrangement of the cylinders in relation to the valve.

The valve employed to distribute the steam to the cylinders is of the piston type, working in a cylindrical steam-chest located in the saddle of the cylinder casting between the cylinders and the smoke-box, and as close to the cylinders as convenience will permit.

As the steam-chest must have the necessary steam passages cast in it and dressed accurately to the required sizes, the main passages in the cylinder casting leading thereto are cast wider than the finished ports. The steam-chest is bored out enough larger than the diameter of the valve to permit the use of a hard cast iron bushing (Fig. 135). This bushing is forced into the steam-chest under such pressure as to prevent the escape of

steam from one steam passage to another except by the action of the valve. Thus an opportunity is given to machine accurately all the various ports, so that the ad-

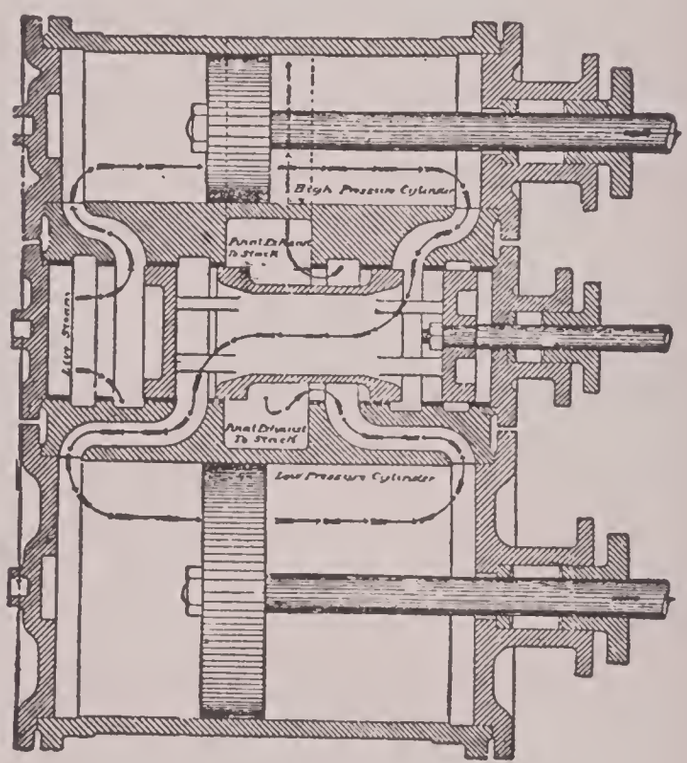


Fig. 134.

mission of steam is uniform under all conditions of service.

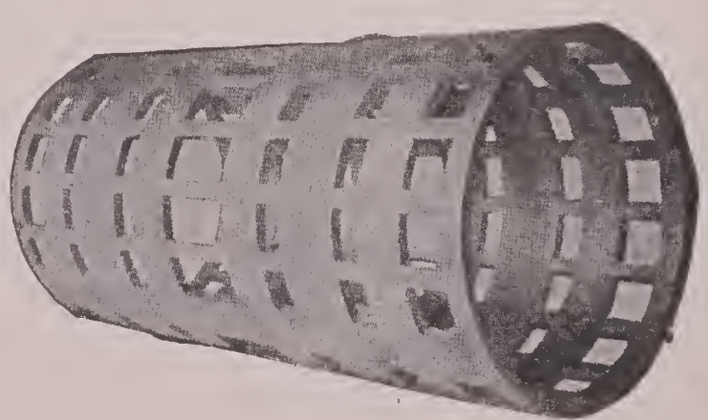


Fig. 135.

The valve, which is of the piston type, double and hollow, as shown in Fig. 136, controls the steam admis-

sion and exhaust of both cylinders. The exhaust steam from the high-pressure cylinder becomes the supply steam for the low-pressure cylinder. As the supply steam for the high pressure cylinder enters the steam-chest at both ends, the valve is in perfect balance, except the slight variation caused by the area of the valve-stem at the back end. This variation is an advantage in case the valve or its connection to the valve-rod should be broken, as it holds them together. Cases are reported where compound locomotives of this system have hauled passenger

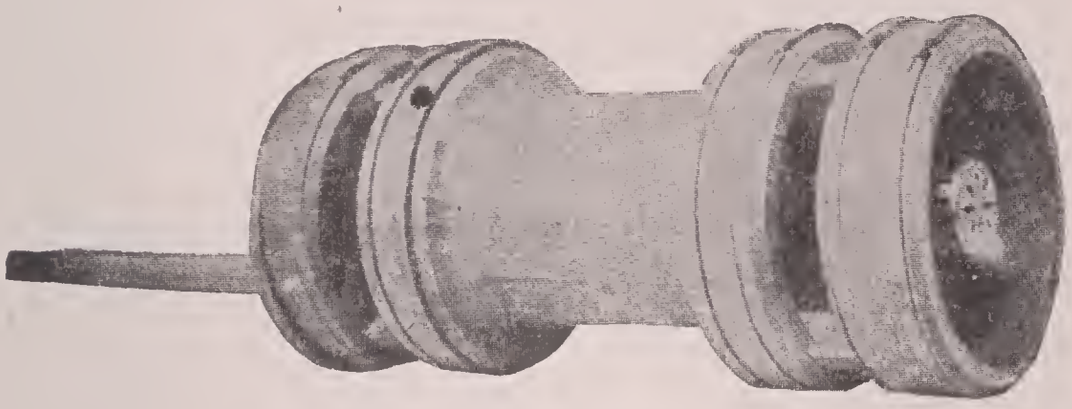


Fig. 136.

trains long distances with broken valve-stems and broken valves, the parts being kept in their proper relation while running by the compression due to the variation mentioned. To avoid the possibility of breaking, it is the present practice to pass the valve-stem through the valve and secure it by a nut on the front end.

Cast iron packing rings are fitted to the valve and constitute the edges of the valve. They are prevented from entering the steam-ports when the valve is in motion by the narrow bridge across the steam-ports of the bushing, as shown in Fig. 135. The operation of the valve is clearly shown in Fig. 134, the direction of the steam being indicated by arrows.

When the low-pressure cylinder is on top, as shown in Fig. 133, the double front rail prevents the use of the ordinary rock-shaft and box, and the valve motion is then what is called "direct acting," changing the location of the eccentrics on the axle in relation to the crank-

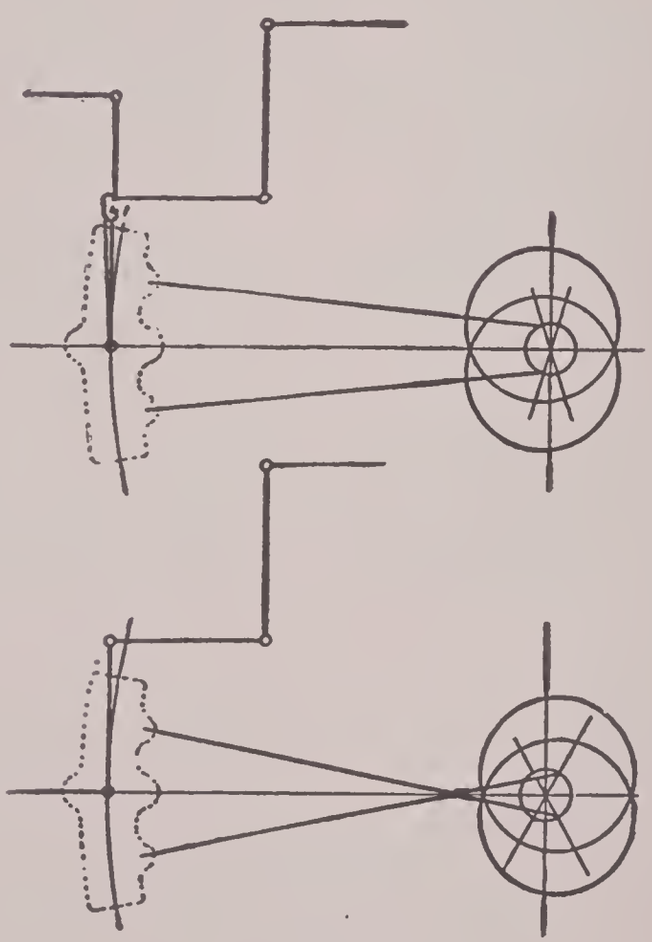


Fig. 137.

pin. When the low-pressure cylinder is underneath, the rocker-shaft is employed, and the eccentrics are placed in the usual position; the valve motion is termed "indirect acting." Fig. 137 shows the relation of the eccentrics with and without the rocker-shaft. Great care should be taken by mechanics, when setting the valves on these loco-

motives, to observe this difference and not get the eccentrics improperly located on the axle. If the crank-pin is placed on the forward center, the eccentric-rods will not be crossed when the rocker-arm or indirect motion is used, but will be crossed when no rocker-arm or direct motion is used. Serious complications have arisen from this being disregarded.

In setting the piston valves, only the high-pressure ports are to be considered. Both heads of the steam chest are removed, and with a tram, from some point on the body of the cylinder to the valve stem, the line and line positions of the valve in both front and back motion, are laid off and indicated by a prick punch mark on the valve stem. Using the same tram, the position of the valve at different parts of the stroke can be ascertained, and the ports noted by the distance from the point of the tram to the prick punch mark. The relation of the low-pressure ports to the valve must be ascertained by measurement, the same as the exhaust ports in ordinary slide valves.

Various methods have been employed to transfer the motion from the links to the valve-rod. That which has proved most satisfactory is to attach the ends of the link and valve-rods to the arms of an intermediate oscillating shaft. This arrangement allows for the free vertical movement of the end of the rod attached to the link, and gives a parallel movement to the valve-rod. It also makes it convenient to obtain any required lateral variation in the line of the two rods. These parts are thoroughly case-hardened, and with reasonable care should wear indefinitely. It is preferable, however, to use a rocker-shaft when possible, as there is then less departure from ordinary locomotive practice.

The cross-head is shown in Fig. 138. It is made of open-hearth cast steel and is machined accurately to size. The bearings for the guide-bars are covered with a thin coating of block tin, about one-sixteenth inch thick, which wears well and prevents heating. The holes for the piston-rods are bored so that the piston-rods will be perfectly parallel, and are tapered to insure a perfect fit.

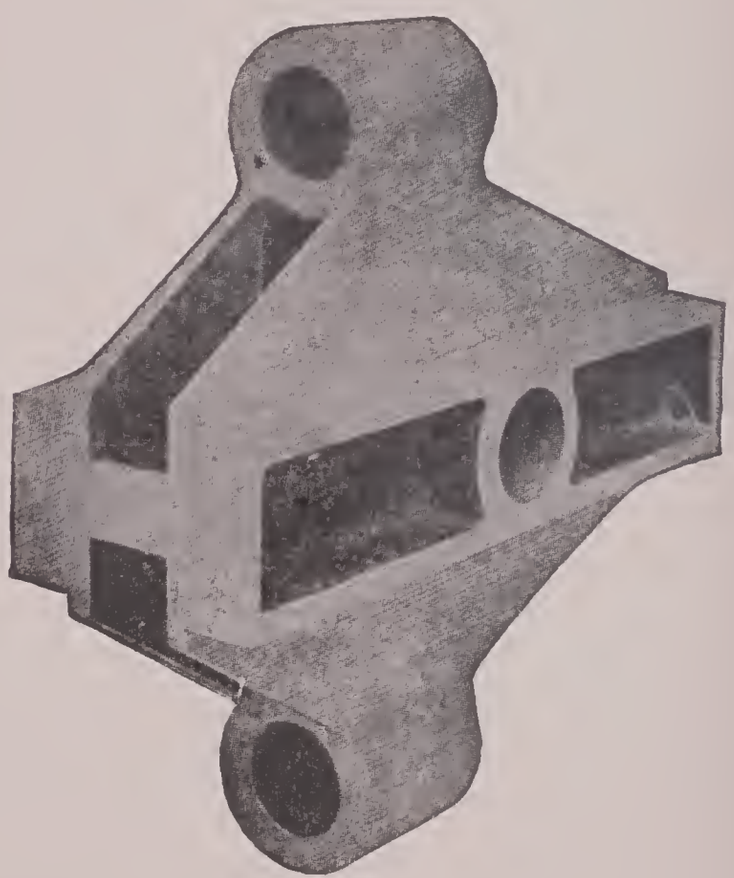
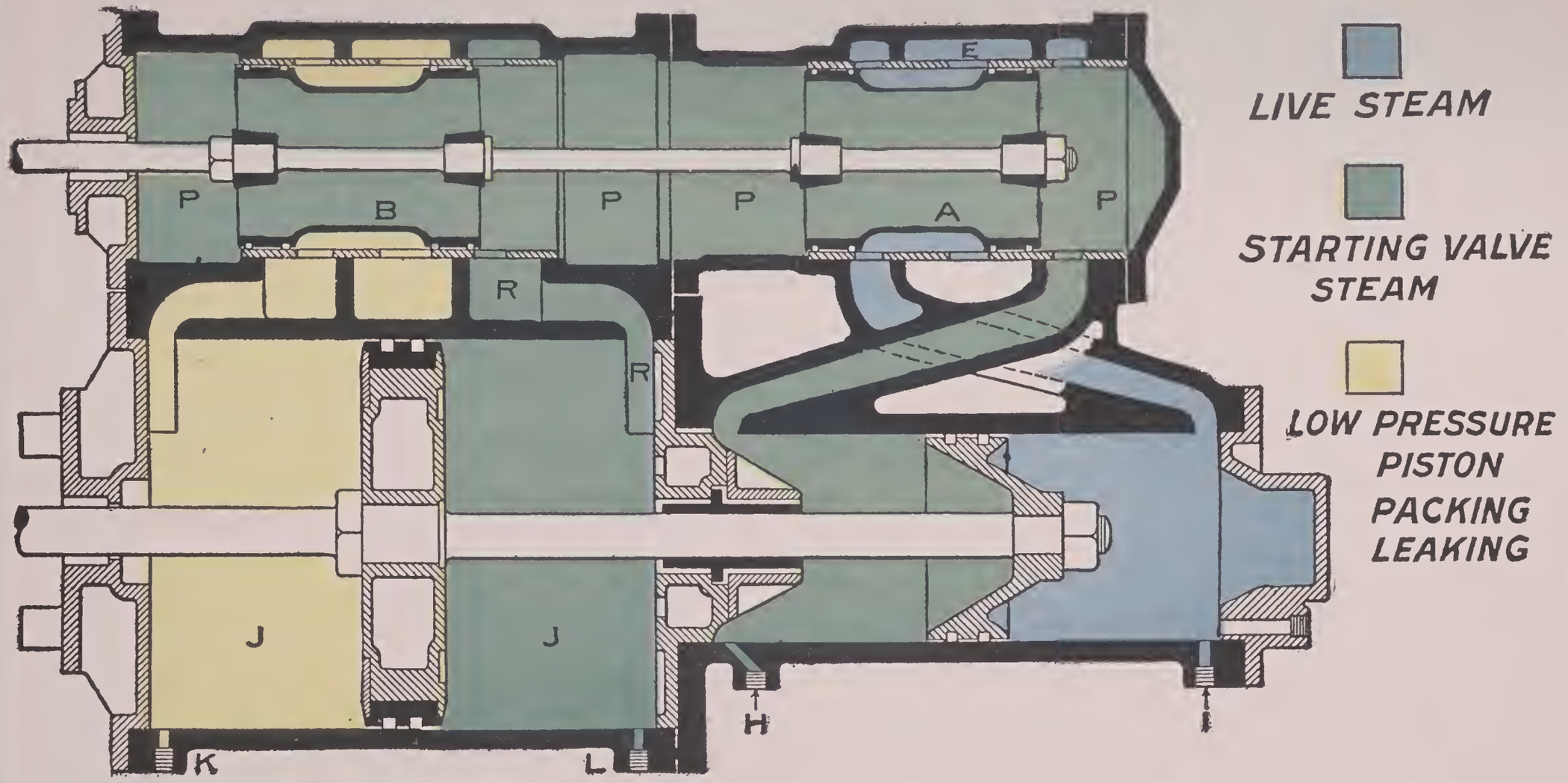


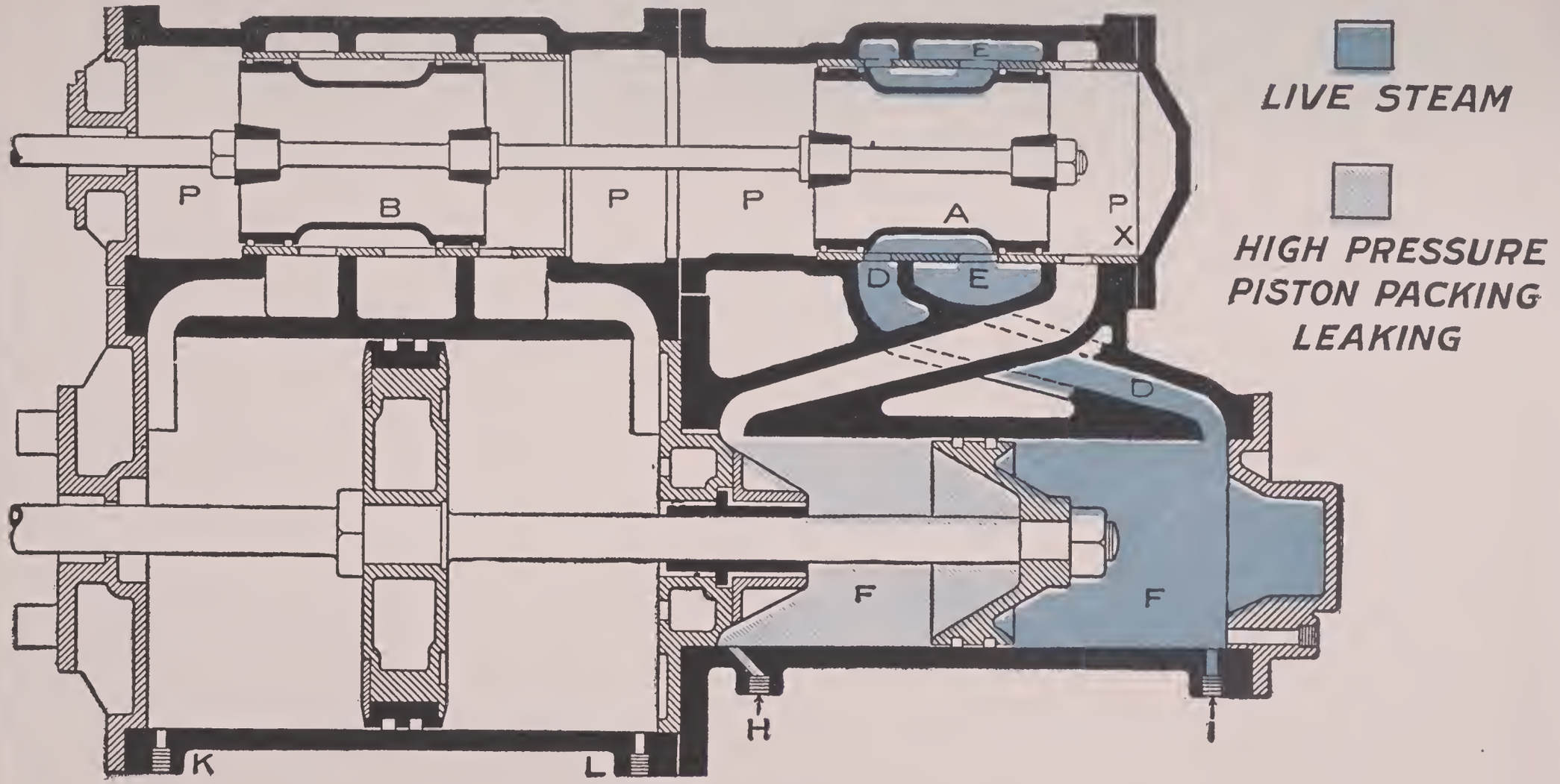
Fig. 138.

The piston shown in Fig. 139 is made with either cast iron or cast steel heads, and is as light as possible. The rods, which are of triple-refined iron, are ground perfectly true to insure good service in connection with metallic packing for the stuffing boxes. The diameter

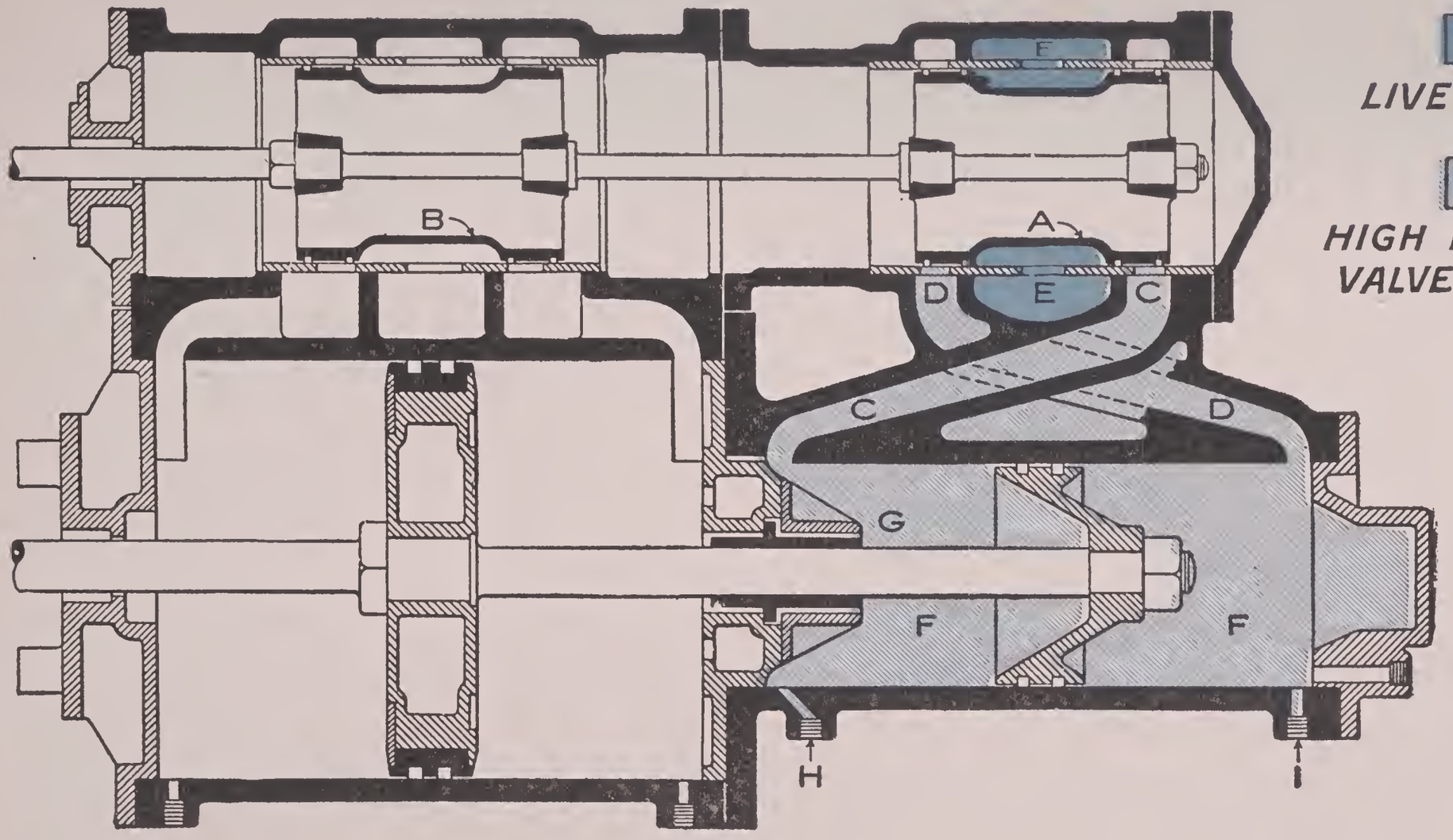
COMPOUND LOCOMOTIVE CYLINDERS SELLERS' INJECTOR



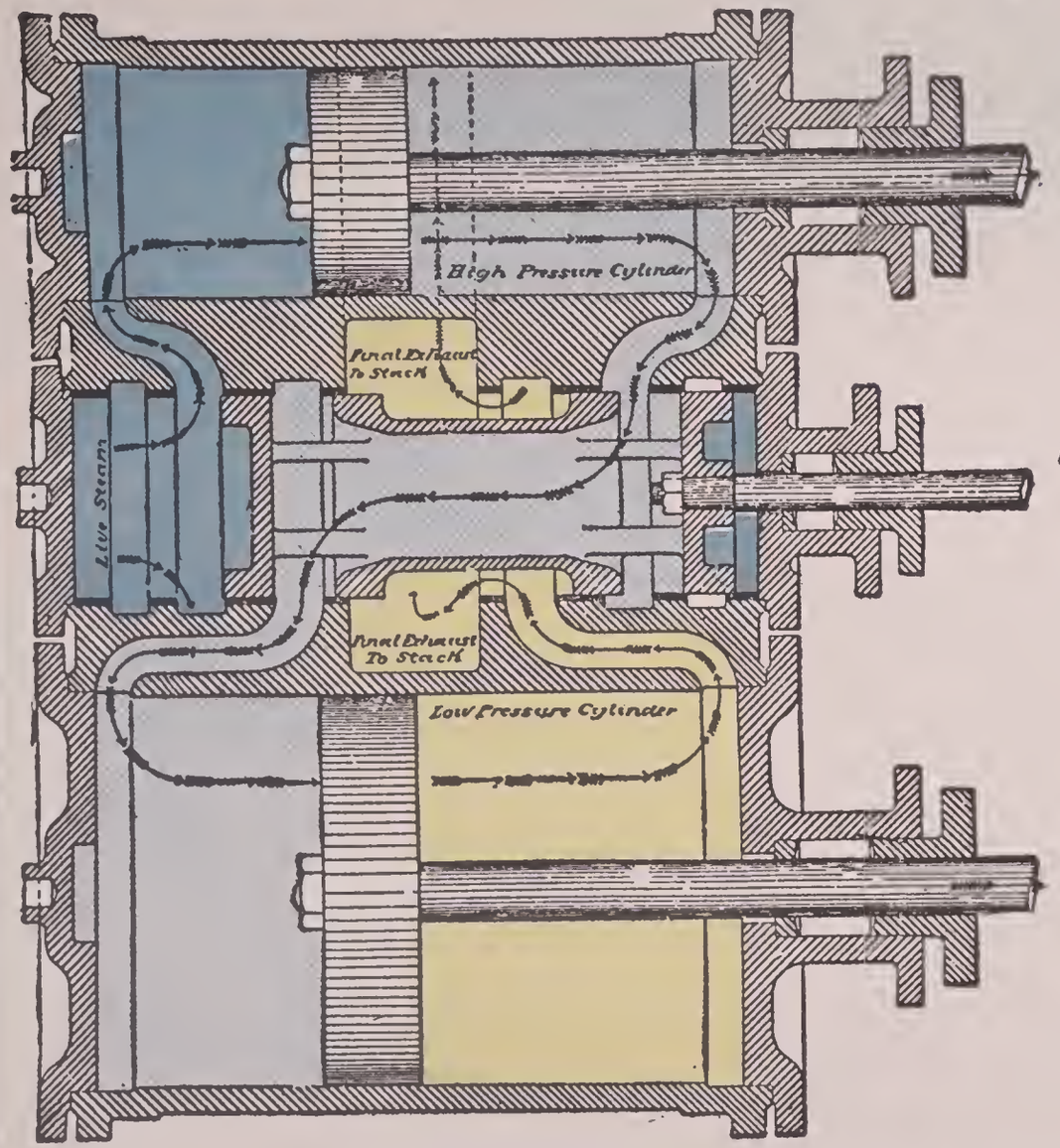
Tandem Compound Locomotive. Testing low pressure cylinder packing.
Engine on top quarter. Reverse lever in back motion. Starting valve open.



Tandem Compound Locomotive.
Testing high pressure piston packing. Engine on top quarter.
Reverse lever in back motion. Starting valve closed.



Tandem Compound Locomotive.
Testing high pressure valve. Both valves in central position.
Engine on top quarter. Reverse lever in center of quadrant.
Throttle slightly open. Starting valve closed.



HIGH PRESSURE CYLINDER

PISTON VALVE

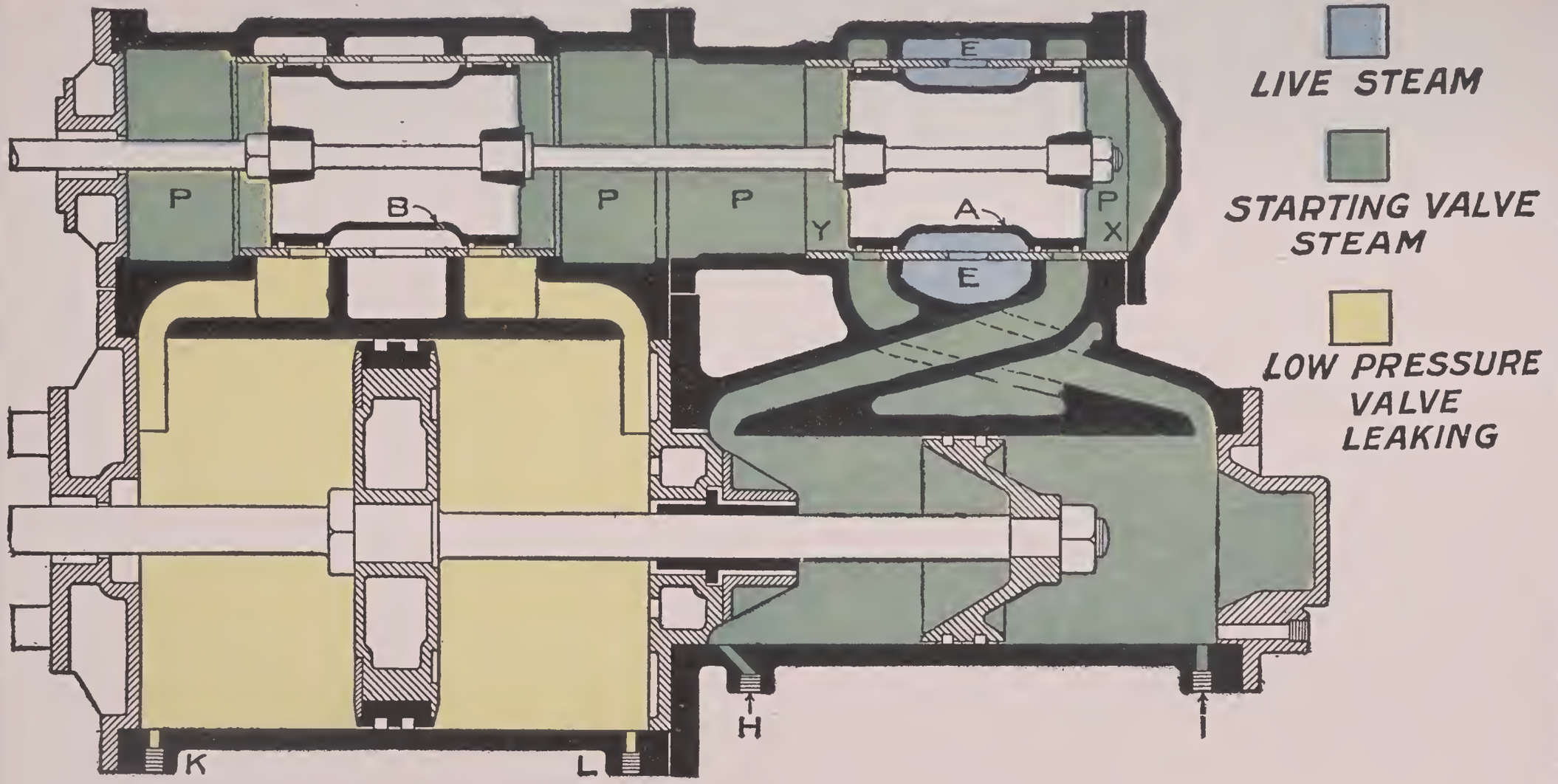
LOW PRESSURE CYLINDER

LIVE STEAM

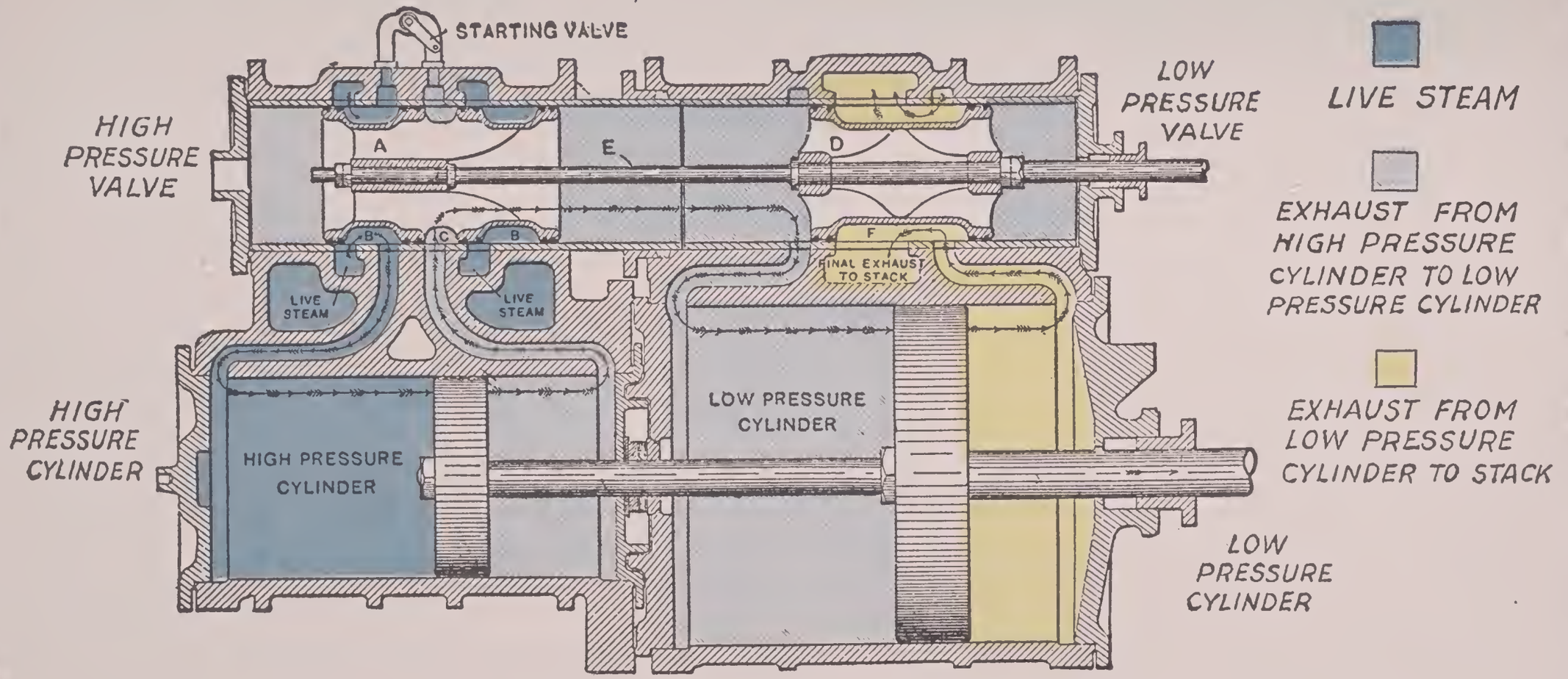
EXHAUST FROM HIGH PRESSURE CYLINDER TO LOW PRESSURE CYLINDER

EXHAUST FROM LOW PRESSURE CYLINDER TO STACK

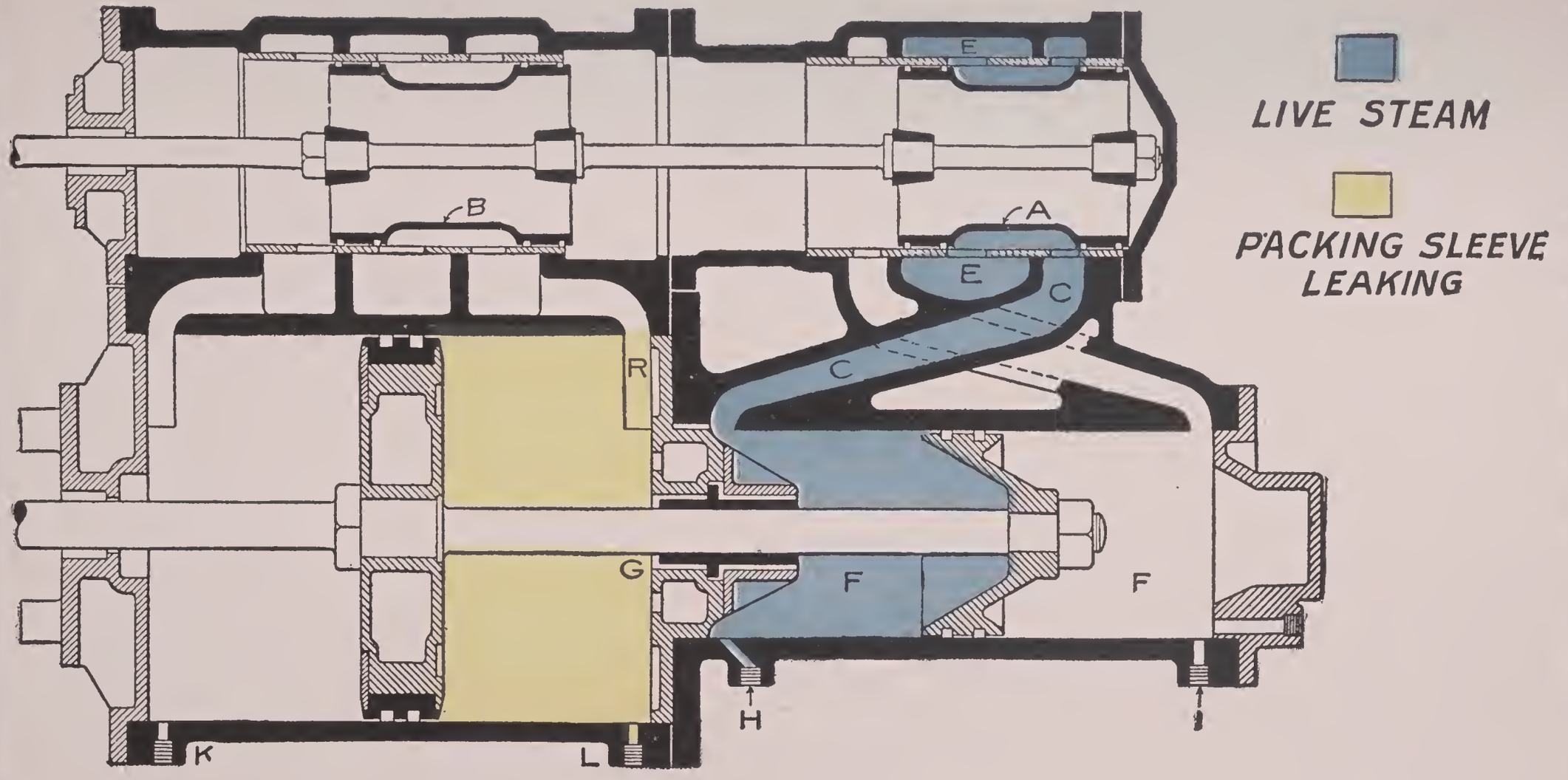
Steam distribution in Vauclain Compound Locomotive.
Built by the Baldwin Locomotive Co.
Pistons nearing half stroke. Vertical section.



Tandem Compound Locomotive. Testing low pressure valve.
Engine on top quarter. Reverse lever in center of quadrant.
By pass valve removed, but valve cap replaced. Starting valve open.



Steam distribution in Tandem Compound Locomotive.
Baldwin Locomotive Co.
Pistons nearing end of back stroke. Vertical section.



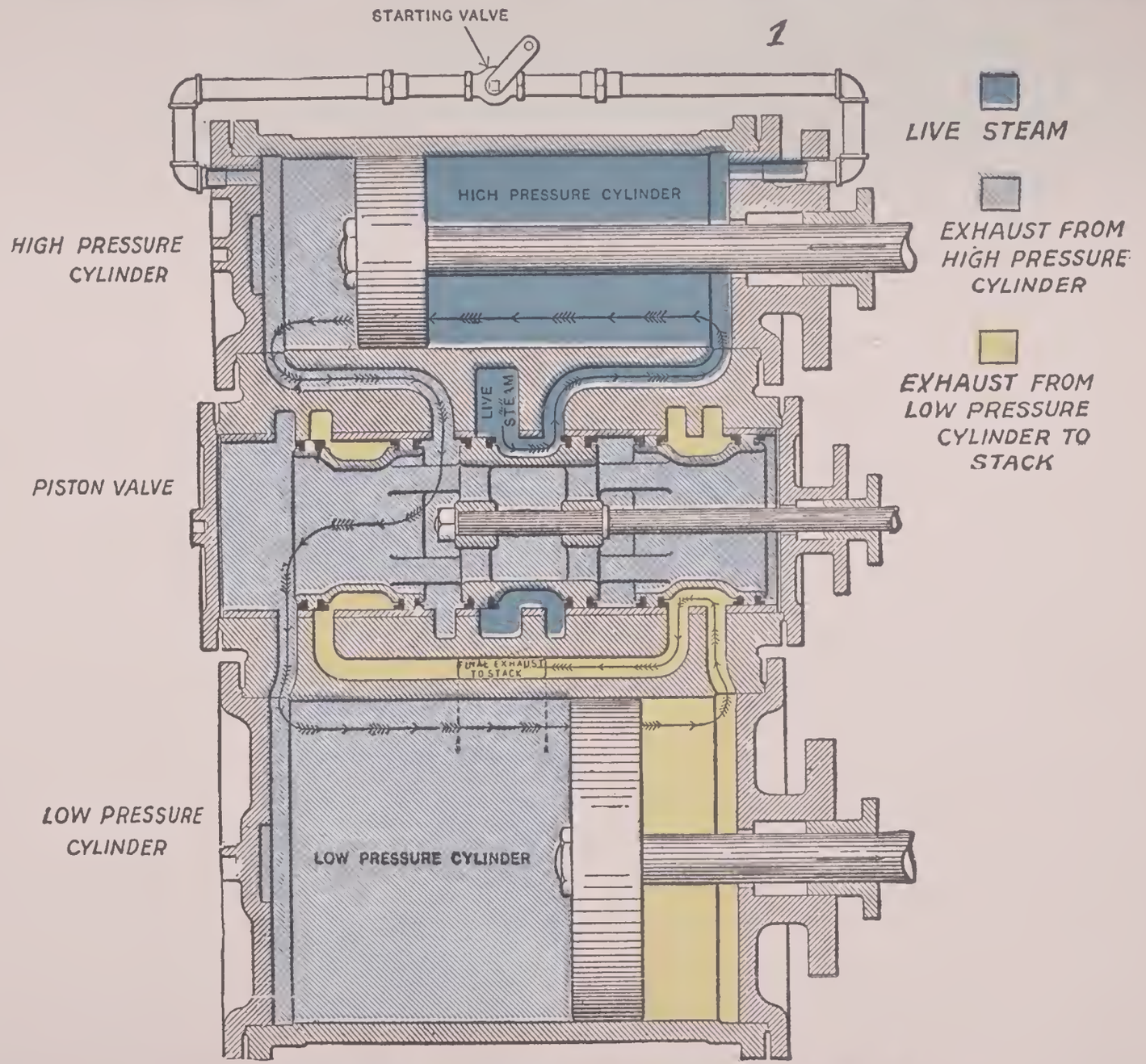
Tandem Compound Locomotive.
Testing packing sleeve between cylinders.
Engine on top quarter. Reverse lever in forward motion. Starting valve closed.

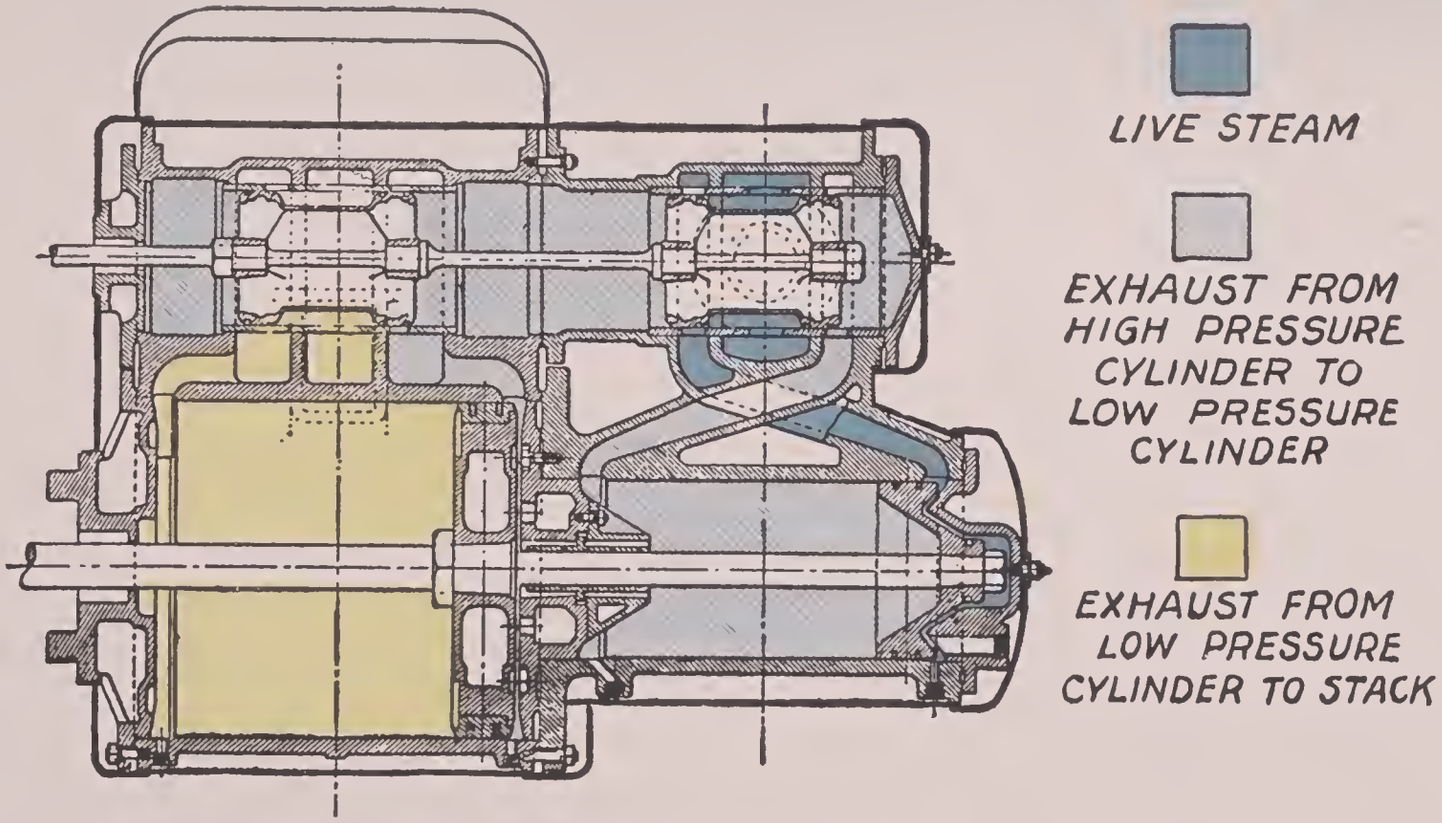
Steam distribution in Four Cylinder Balanced Compound Locomotive, built by the Baldwin Locomotive Company.

Plan view of cylinders and valves.

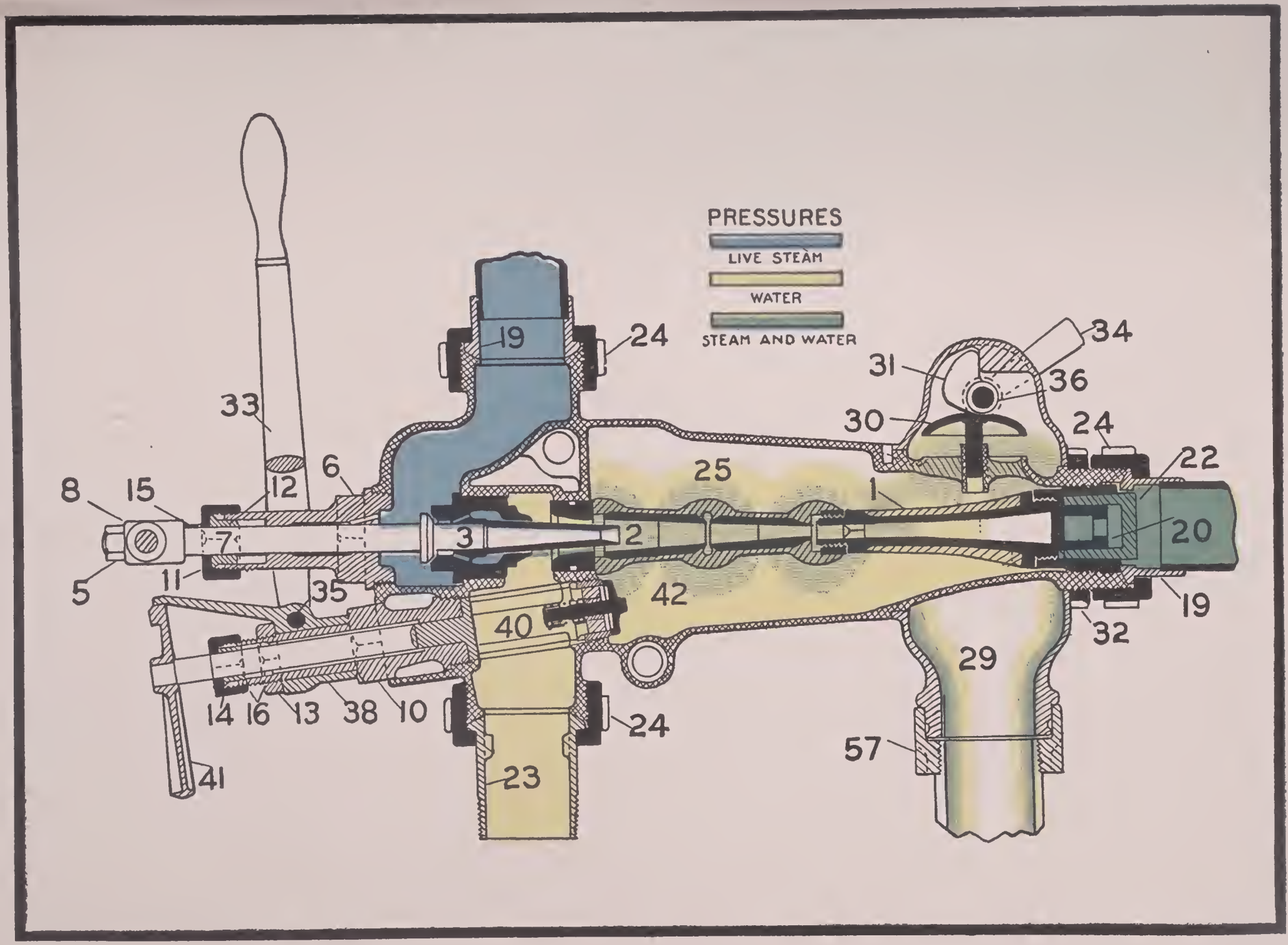
High pressure piston on forward stroke.

Low pressure piston on return stroke.

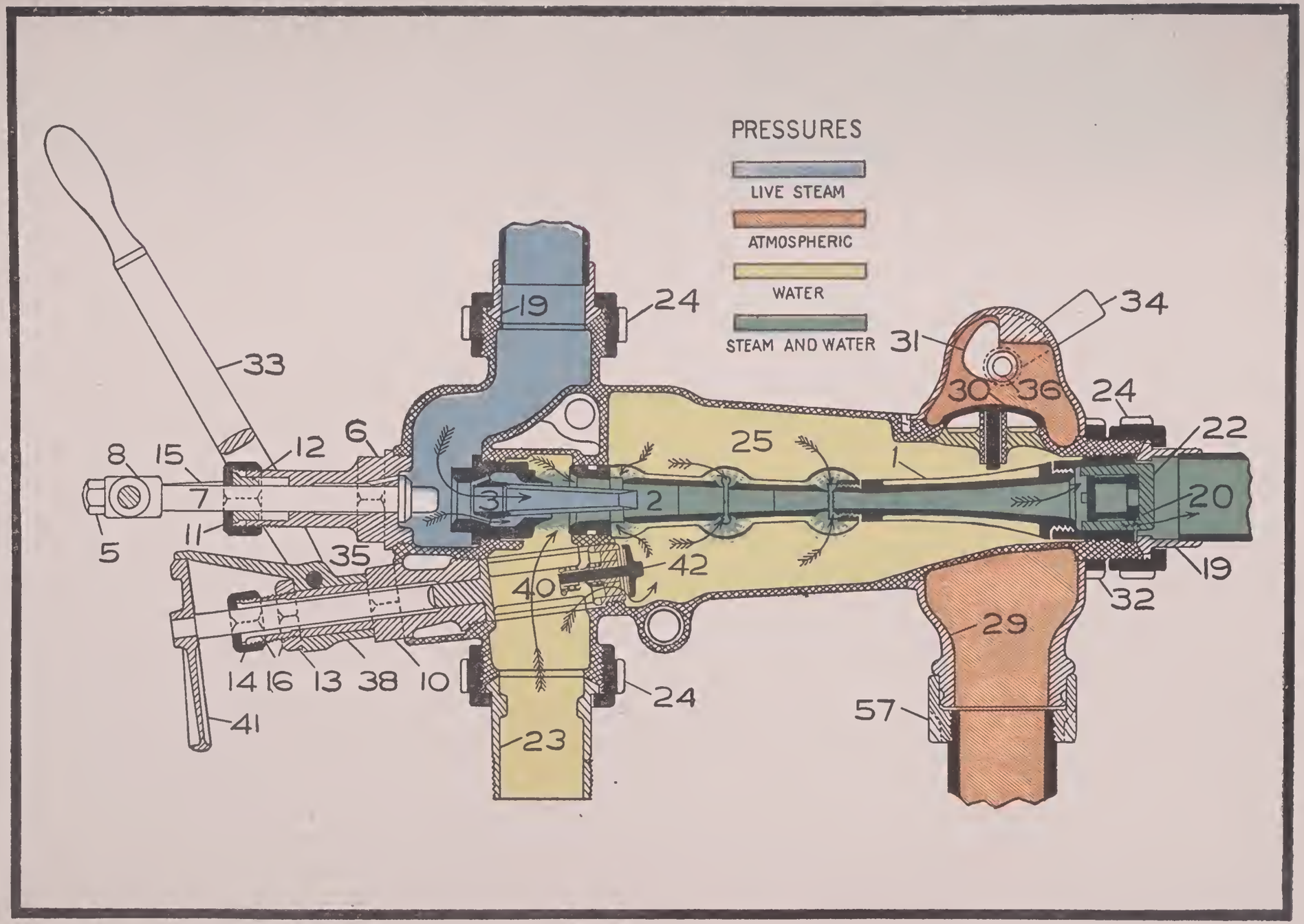




Steam distribution in Tandem Compound Locomotive.
Built by American Locomotive Co.
Pistons beginning back stroke. Valves admitting steam. Vertical section.



Locomotive Firemen's Magazine. Sellers' Improved Class "N" Self-Acting Injector.—(Priming position). Boiler operation.



Locomotive Firemen's Magazine. Sellers' Improved Class "N" Self-Acting Injector.—(Working position). Boiler operation.

of both piston-rods is the same, both having equal work to perform. They are made large enough to resist strains due to any unequal pressure that may come upon them in starting the locomotive from a state of rest. The cross-head end has a shoulder which prevents the piston-

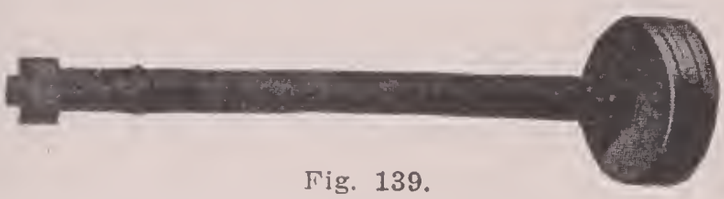


Fig. 139.

rod being forced into the cross-head, and at the same time permits the cross-head end and the body of the piston-rod to be of one diameter, thus permitting vibratory strains to act throughout the entire length of the rod instead of concentrating them at the shoulder next to the cross-head. The piston-rods are secured to the cross-head by large nuts, and these in turn are prevented from coming loose by taper keys driven tightly against them.

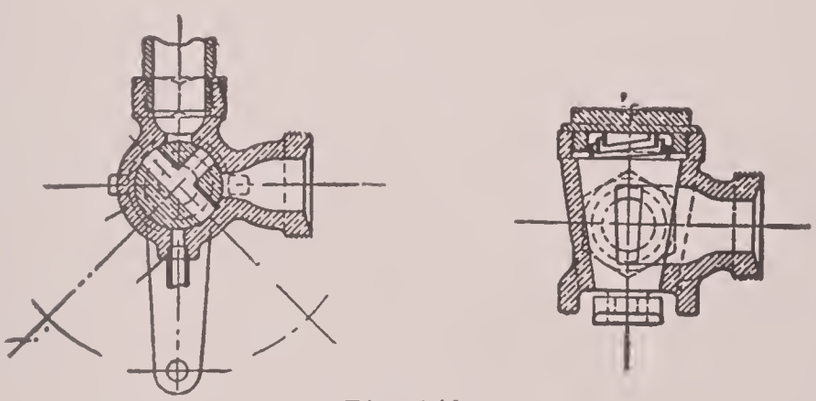


Fig. 140.

It is obvious that in starting these locomotives with full trains from a state of rest, it is necessary to admit steam to the low-pressure cylinder as well as to the high-pressure cylinder, which is accomplished by the use of a starting valve (Fig. 140). This is merely a pass-by valve

which is opened to admit steam to pass from one end of the high-pressure cylinder to the other end, and thence through the exhaust to the low-pressure cylinder. This is more clearly shown at E in Figs. 142 and 143. The same cock acts as a cylinder cock for the high-pressure cylinder and is operated by the same lever that operates the ordinary cylinder cocks, thus making a simple and efficient device and one that need not become disarranged. This valve should be kept shut as much as possible, as its indiscriminate use reduces the economy and makes the locomotive "logy."

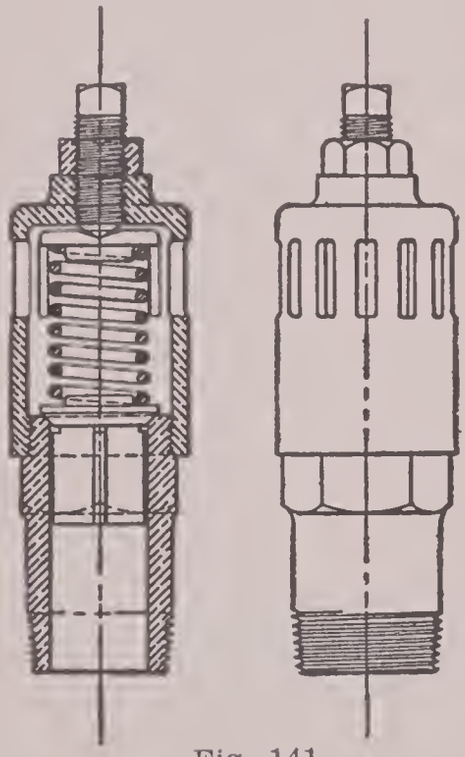


Fig. 141.

As is usual in all engines, air valves are placed in the main steam passage of the high-pressure cylinder. Additional air valves, marked C and C^1 in Fig. 142, are placed in the steam passages of the low-pressure cylinders to supply them with sufficient air to prevent the formation of a vacuum which would draw cinders into the steam chest and cylinders.

The hollow valve stem shown in Fig. 143 accomplishes the same result, but with a more direct action, and is preferable for fast service. The check valve at the end of the hollow stem outside the steam chest is closed by the pressure of the steam, but stands open when the pressure is relieved and air is allowed to pass into the valve through the perforation in the hollow stem. A vacuum is thus prevented from forming in the valve or low-pressure

passages. This arrangement will also prevent the accidental starting of the locomotive occasioned by a leaky throttle. The steam as it slowly escapes will pass through the hollow stem to the open air without creating pressure in the cylinders.

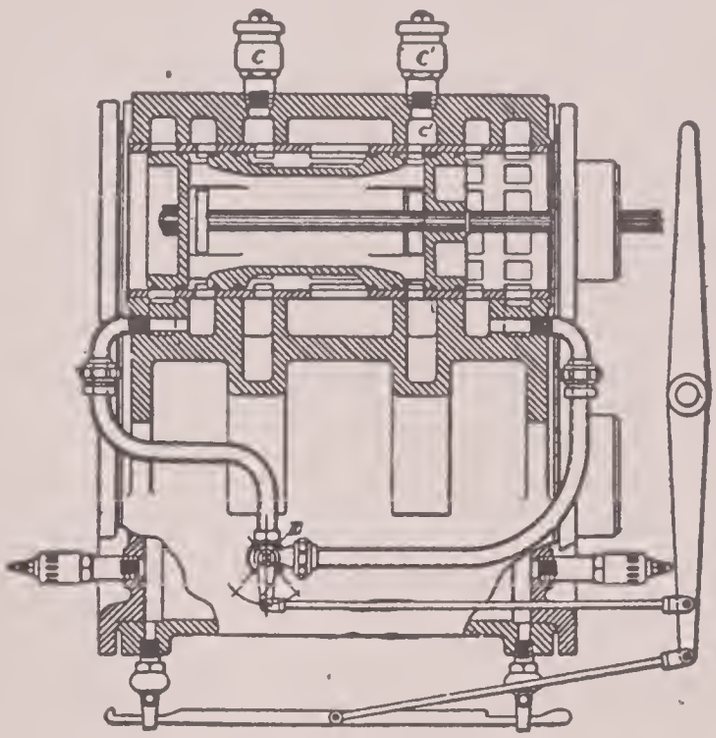


Fig. 142.

Water relief valves (Fig. 141) are applied to the low-pressure cylinders and attached to the front and back cylinder heads to prevent the rupture of the cylinder in case a careless engineer should permit the cylinders to be charged with water, or to relieve excessive pressure of any kind.

In all other respects the locomotive is the same as the ordinary single-expansion locomotive.

Operation. It is not surprising, in view of their differences of opinion respecting single-expansion locomotives, that there has been much controversy among engi-

neers and firemen in regard to the operation of compound locomotives of this system. The first thing the engineer must learn is to use the reverse lever for what it is intended; that is, he must not hesitate to move it forward when ascending a grade if the locomotive shows signs of slowing up. The reverse quadrant is always so made that it is impossible to cut off steam in the high-pressure cylinder at less than half stroke, which avoids the damage that might ensue from excessive compression.

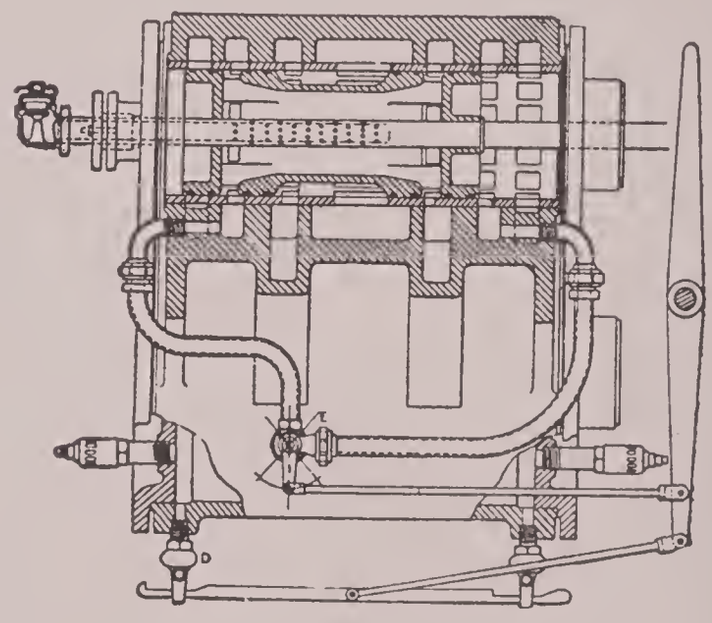


Fig. 143.

It is perfectly practicable to operate the engine at any position of the reverse lever between half stroke and full stroke, without serious injury to the fire. When starting the locomotive from a state of rest, the engineer should always open the cylinder cocks to relieve the cylinders of condensation, and as the starting valve is attached to the cylinder cocks, this movement also admits steam to the low-pressure cylinder and enables the locomotive to start quickly and freely. In case the locomotive is attached to

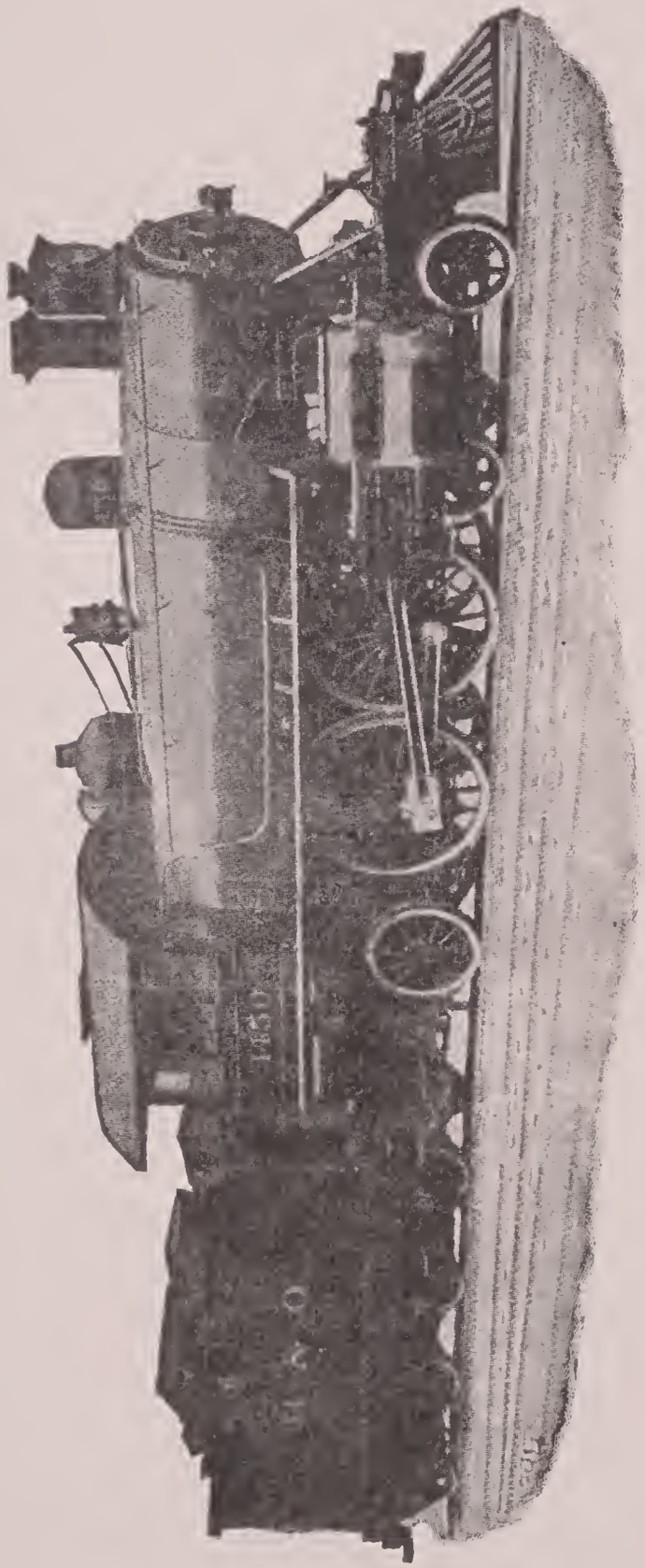


Fig. 144. Atlantic Type Locomotive, Baltimore & Ohio Railroad.

a passenger train and standing in a crowded station, or in some position where it is undesirable to open the cylinder cocks, the engineer should move the cylinder cock lever in position to permit live steam to pass by into the low-pressure cylinder, thus enabling the locomotive to start quickly and uniformly, without any of the jerking motion so common in two-cylinder or cross-compound locomotives. After a few revolutions have been made and the cylinders are free from water caused by condensation or priming, the engineer should move the cylinder cock lever into the central position, causing the engine to work compound entirely. This should be done before the reverse lever is disturbed from its full gear position. The reverse lever should never be "hooked up," thereby shortening the travel of the valve, until after the cylinder cock lever has been placed in the central position. It is often necessary to open the cylinder cocks when at full speed, to allow water to escape from the cylinders, especially when the engineer is what is commonly called a "high-water" man, and in such case no disadvantage is experienced and the reverse lever need not be disturbed. The starting device should not be used for any other purpose than the "starting" of the train. After the train is in motion it should not be used. Cases have been observed where the engineers use it all the time and have the reverse lever "hooked up" in the top notch (half stroke), in consequence of which the locomotive will slow down to a low speed whilst burning an excessive amount of coal. Such running must result in general dissatisfaction.

The starting device is useful in emergencies, as, for instance, when stalling with a heavy train on a grade, if live steam is admitted to the low-pressure cylinder

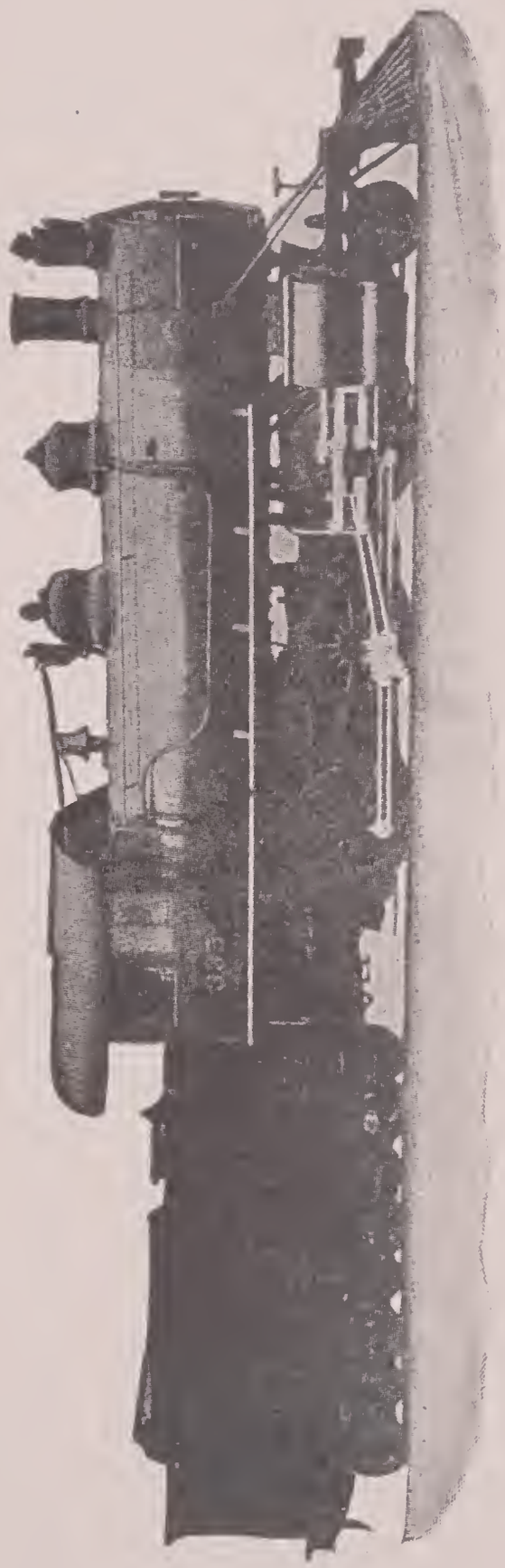


Fig. 145. Mogul Freight Locomotive, Chicago & Alton Railway.

sufficient additional power is obtained to start the train and take it over the grade. This should be resorted to only in emergencies, and allowance should be made for the extra repairs caused by frequent cases of this kind.

On account of the very mild exhaust, the fireman should carry the fire as light as possible. A little practice will enable him to judge how to get along with the least amount of fuel.

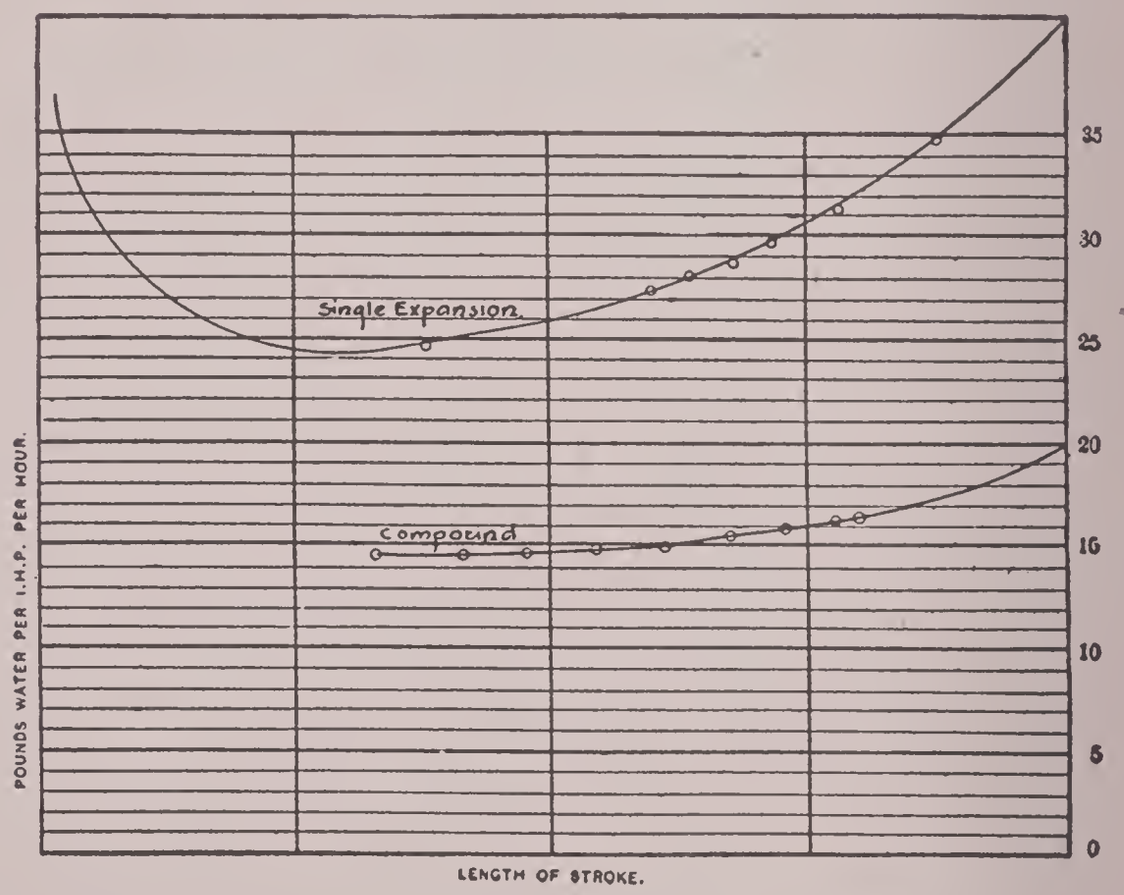


Fig. 146.

The diagram (Fig. 146) shows the difference in the amount of water required to do the work at various points of cut-off in compound and single-expansion locomotives. The upper line shows the rate of water consumption per horse-power developed for several points of cut-off in single-expansion locomotives, whilst the

lower line shows the same for compound locomotives. It will be observed that the most economical point of cut-off is about one-quarter stroke on the single-expansion locomotive, and about five-eighths stroke on the compound locomotive. It is also noticeable that the water-rate per horse-power varies very little on the compound locomotive when the reverse lever is moved towards full gear or longer cut-off, but in the single-expansion engine it increases rapidly, causing engineers to remark that they cannot "drop her a notch" on account of "getting away with the water." This does not occur with the compound locomotive when the reverse lever is moved forward towards full gear, and no engineer should open the pass-by valve, admitting live steam to the low-pressure cylinder, until the last notch has been used on the quadrant and the engine is about to stall.

It is also desirable to move the reverse forward a notch before the locomotive slows down too much, as it is better to preserve the momentum of the train than to slow down and again have the trouble of accelerating. In this way both coal and water are wasted. If these instructions are observed the locomotive will work satisfactorily.

Repairs. On account of the great similarity to single-expansion locomotives, mechanics familiar with the latter have no difficulty in understanding these compound locomotives. There is no new element of repairs introduced, no complicated starting or reducing valves, such as are common to other systems of compound locomotives.

The cross-heads, when badly worn, may, in a short time, be retinned by any coppersmith; in fact, an ordinary laborer can be taught this in a few days. The cross-head is heated warm enough to melt solder, and is then

cleaned and wiped with solder, using dilute muriatic acid, such as tinsmiths use in soldering. Block tin is then poured against the surfaces so prepared, to which it adheres. A piece of iron placed alongside the cross-head can be used to regulate the thickness.

The cross head is then put on a planer to true it up, care being used not to let the tool "dig in" and tear off the tin.

The pistons are treated the same as in ordinary single-expansion engines. The packing-rings in the low-pressure cylinder require renewal more frequently than those in high-pressure cylinders. It is also more difficult in compound cylinders to detect faulty packing rings, and they are sometimes noticed only by the locomotive failing in steam and in not making time on the road.

The piston-valves should last a long time if properly lubricated, but when the bushing (Fig. 135) and valve (Fig. 136) are worn enough to require attention, the bushing should be bored out and new rings put in the valve; very often it is not necessary to bore the bushings, merely to put new packing-rings in the valve.

After the bushings (Fig. 135) have been bored several times, larger valves may be fitted to them, so as to have as little play as possible. A very convenient type of boring bar for boring out the bushings has been designed, by which the work can be done without taking down the back head of the steam-chest. It is possible with this tool to bore out the bushings in less time than required to face a valve seat on a single-expansion locomotive.

When putting new bushings in the steam-chests, the device shown in Fig. 147 may be used, which gives the required power and is slow enough to permit the bushing to accommodate itself to the cylinder casting.

When extracting old bushings, it is best to split them with a narrow cape chisel—they are only fit for scrap when removed, and can be much more quickly removed this way than to attempt to draw them out with draw screws.

Enough attention should be given the starting valves to insure their moving in harmony with each other. Engineers sometimes strain the cylinder cock shaft, which causes one starting valve to open and the other to remain shut; this causes the exhaust to beat unevenly, and the engineer is apt to complain that the valves are out of

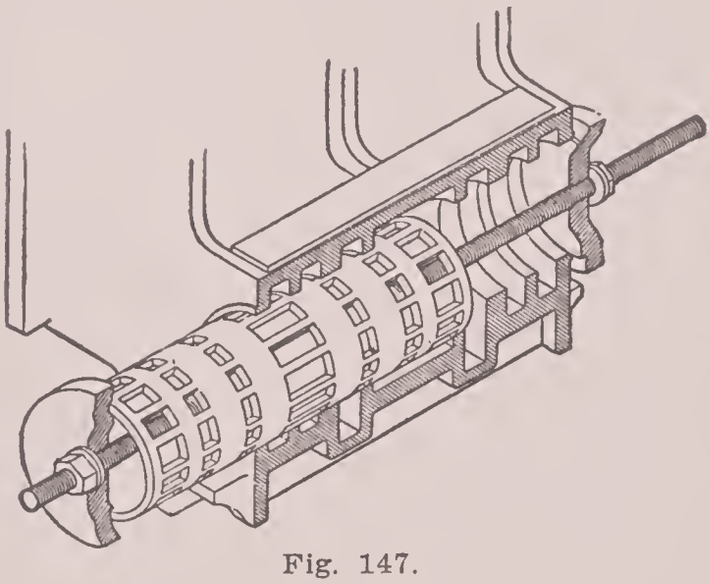


Fig. 147.

square. Before altering the valve motion on these engines, make sure that the starting valves open and close simultaneously, and examine low-pressure pistons and piston-valve for broken packing-rings. In one case an engineer ran his locomotive two days without any piston-head on one of the low-pressure pistons, and even then could not tell what was the matter, only that the locomotive sounded "lame" and did not make good time with the train. Men were put to work to locate the trouble, and found it to the great surprise of the engineer.

Suggestions for Running a Vauclain Four-Cylinder Compound Locomotive. In starting the locomotive with a train, place the reverse lever in full forward position, throw the cylinder-cock lever forward, which operation opens the starting-valve and allows live steam to pass to the low-pressure cylinder. The throttle is then opened, and as soon as possible when the cylinders are free of water and the train is under good headway, the cylinder cocks and starting-valve should be closed. As the economy of a compound locomotive depends largely on its greater range of expansion, the engineer should bear in mind that in order to get the best results he must use his reverse lever. After the starting-valve is closed and as the speed of the train increases, the reverse lever should be hooked back a few notches at a time until the full power of the locomotive is developed. If after moving the reverse lever to the last notch, which cuts off the steam at about half stroke in the high-pressure cylinder, it is found that the locomotive develops more power than is required, the throttle must be partially closed and the flow of steam to the cylinder reduced. On slightly descending grades the steam may be throttled very close, allowing just enough in the cylinders to keep the air-valves closed. If the descent is such as to prevent the use of steam, close the throttle and move the reverse lever gradually to the forward notch and move the starting-lever to its full backward position. This allows the air to circulate either way through the starting-valve from one side of the piston to the other, relieves the vacuum, and prevents the oil from being blown out of the cylinder. On ascending grades with heavy loads as the speed decreases the reverse lever should be moved forward sufficiently to keep up the required speed. If,

Fig. 148. Consolidation Locomotive, Lehigh Valley Railroad.

after the reverse lever is placed in the full forward notch, the speed still decreases and there is danger of stalling, the starting-valve may be used, admitting steam to the low-pressure cylinders. This should be done only in cases of emergency and the valve closed as soon as the difficulty is overcome.

The tractive power of Vauclain four-cylinder compound locomotives may be ascertained by the following formula:

$$\frac{C^2 \times S \times \frac{2}{3} P}{D} + \frac{c^2 \times S \times \frac{1}{4} P}{D} = T, \text{ in which}$$

C = Diameter of high-pressure cylinder in inches.
c = Diameter of low-pressure cylinder in inches.
S = Stroke of piston in inches.
P = Boiler pressure in pounds.
D = Diameter of driving wheels in inches.
T = Tractive power.

It is not claimed for compound locomotives that a heavier train can be hauled at a given speed than with a single-expansion locomotive of similar weight and class. No locomotive can haul more than its adhesion will allow; but the compound will, at very slow speed on heavy grades, keep a train moving where a single-expansion locomotive will slip and stall. This is due to the pressure on the crank-pins of the compound being more uniform throughout the stroke than is the case with the single-expansion locomotive.

The principal object in compounding locomotives is to effect fuel economy, and this economy is obtained—

1. By the consumption of a smaller quantity of steam in the cylinders than is necessary for a single-expansion locomotive doing the same work.

2. The amount of water evaporated in doing the same work being less in the compound, a slower rate of combustion combined with a mild exhaust produces a higher efficiency from the coal burned.

In a stationary engine, which does not produces its own steam supply, it is of course proper to measure its efficiency solely by its economical consumption of steam. In an engine of this description the boilers are fired independently, and the draft is formed from causes entirely separate and beyond the control of the escape of steam from the cylinders; hence, any economy shown by the boilers must of necessity be separate and distinct from that which may be effected by the engine itself. In a locomotive, however, the amount of work depends entirely upon the weight on the driving-wheels, the cylinder dimensions being proportioned to this weight; and whether the locomotive is compound or single-expansion, no larger boiler can be provided, after allowing for the wheels, frames, and other mechanism, than this weight permits. Therefore, the heating surface and grate area are practically the same in both types, and the evaporative efficiency of both locomotives is determined by the action of the exhaust, which must be of sufficient intensity in both cases to generate the amount of steam necessary for utilizing, to the best advantage, the weight on the driving-wheels. This is a feature that does not appear in a stationary engine, so that the compound locomotive cannot be judged by stationary standards, and the only true comparison to be made is between locomotives of similar construction and weight, equipped in one case with compound and in the other with single-expansion cylinders.

One of the legitimate advantages of the compound system is that, owing to the better utilization of the steam,

less demand is made upon the boiler, which enables sufficient steam-pressure to be maintained with the mild exhaust, due to the low tension of the steam when exhausted from the cylinders. This milder exhaust does not tear the fire, nor carry unconsumed fuel through the flues into the smoke-box and thence out of the smoke-stack, but is sufficient to maintain the necessary rate of combustion in the fire-box with a decreased velocity of the products of combustion through the flues.

The heating surfaces of a boiler absorb heat units from the fire and deliver them to the water at a certain rate. If the rate at which the products of combustion are carried away exceeds the capacity of the heating surfaces to absorb and deliver the heat to the water in the boiler, there is a continual waste that can be overcome only by reducing the velocity of the products of combustion passing through the tubes. This is effected by the compound principle. It gives, therefore, not only the economy due to a smaller consumption of water for the same work, but the additional economy due to slower combustion. It is obvious that these two sources of economy are interdependent.

The improved action of the boiler can be obtained only by the use of the compound principle, while the use of the compound principle enables the locomotive to develop its full efficiency under conditions which in a single-expansion locomotive would require a boiler of capacity so large as to be out of the question under the circumstances usually governing locomotive construction. It is therefore evident that where both locomotives are exact duplicates in all their parts, excepting the cylinders, the improved action of the boiler is due entirely to the compound principle, and the percentage of economy

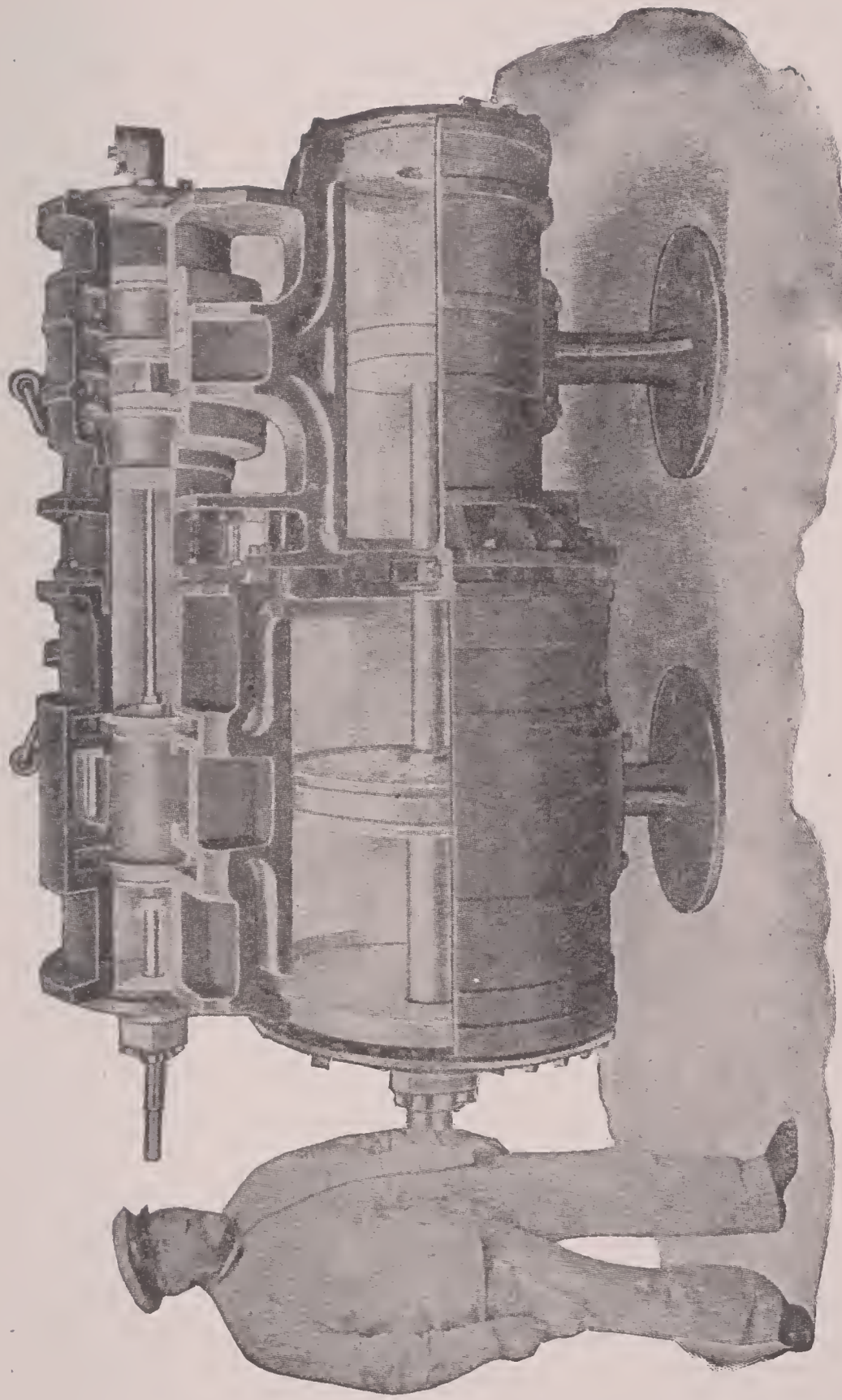


Fig. 149. Model of Actual Size, Showing Two Locomotive Cylinders and Valves of the Vauclain Tandem Compound System.

should be based upon the total saving in fuel consumption, and not upon water consumption, as in stationary practice.

For the benefit of those who may test these locomotives, the following method is presented, of determining the water rate per horse-power from an indicator diagram:

S = Stroke in inches.
C = Per cent of stroke completed at cut-off.
P = Pressure of steam at cut-off, taken from zero.
WP = Weight per cubic foot of steam at P pressure.
H = Per cent of stroke uncompleted at compression.
Q = Pressure of steam at compression taken from zero.
Wq = Weight per cubic foot of steam at Q pressure.
E = Per cent of clearance in H.-P. cylinders.
A = Area of H.-P. cylinders.
P = M.E.P. of H.-P. cylinders.
a = Area of L.-P. cylinders.
K = M.E.P. of L.-P. cylinders.
N = Number of revolutions per minute.
r = Ratio $\frac{a}{A}$; hence, $a = A \times r$.

All calculations are made on the basis of the high-pressure cylinder doing the work of both cylinders.

The volume of the piston displacement is AS, and the volume at cut-off is $A\ S\ C$, since C is the proportion of stroke completed at cut-off. The volume of N revolutions would be $A\ N\ S\ C$. As there are two strokes of the piston for each revolution, and there is an engine on each side of the locomotive, assuming that both engines are doing exactly the same work, there would be four strokes per revolution; hence $4\ A\ N\ S\ C$ is the volume of piston displacement at cut-off for one revolution.

Since the clearance-space is expressed in percentage of the piston displacement of one stroke, and this space is filled at each stroke, the volume of the clearance-space for one revolution would be $4\ A\ N\ S\ E$. The sum of these two quantities divided by 1,728 will give the volume in cubic feet. The indicator-card gives the pressure at cut-off, and a reference to Table 4 will give the weight of steam at that pressure; hence, the amount of steam used per revolution becomes $\left(\frac{4\ A\ N\ S\ C + 4\ A\ N\ S\ E}{1{,}728}\right) Wp$. But there is a certain amount of steam saved at compression, and the volume at this point would be $\left(\frac{4\ A\ N\ S\ H + 4\ A\ N\ S\ E}{1{,}728}\right) Wq$, the volume of the clearance space being again taken into consideration. Since this steam is saved by compression, it should be deducted from the amount used, and the formula becomes:

$$\left(\frac{4ANSC+4ANSE}{1{,}728}\right) Wp - \left(\frac{4ANSH+4ANSE}{1{,}728}\right) Wq;$$

or $$\frac{4\ A\ N\ S}{1{,}728}\Big(\ (C + E)\ Wp - (H + E)\ Wq.\ \Big).$$

The H.-P. equals $\frac{4\ A\ N\ S\ (P + r\ K)}{12 \times 33{,}000}$

Then the water rate per minute would be

$$\frac{\frac{4\ A\ N\ S}{1{,}728}\Big(\ (C + E)\ \ Wp - (H + E)\ Wq\ \Big)}{\frac{4\ A\ N\ S\ (P + r\ K)}{12 \times 33{,}000}},$$

or $$\frac{229.16}{P + r\ K}\Big(\ (C + E)\ Wp - (H + E)\ Wq\ \Big);$$

and the rate per hour would be $\frac{60 \times 229.16}{P + r\,K}$,

or $\frac{13{,}750}{P + r\,K}\left((C + E)\cdot Wp - (H + E)\ Wq \right)$, which formula is to be used.

If it is desired to get the steam at release H.-P., substitute the value of the point R and pressure t, also $S \times R$, respectively, for C, p, and $C \times S$. See Figs. 150 and 151.

M.E.P. H.-P. cylinder..87 pounds Clearance.. .08
M.E.P. L.-P. cylinder...32 pounds Ratio..2.87 to 1
M.E.P. referred to H.-P. cylinder.....178.84
M.E.P. referred to L.-P. cylinder..... 62.31

$$178.84 = P + r\,K$$
$$62.31 = K + \frac{P}{r}$$

$\frac{135.3}{150.0} = .3376$ pound per cubic foot of steam at cut-off H.-P. cylinder.

$\frac{60.3}{75.0} = .1756$ pound per cubic foot of steam at compression H.-P. cylinder.

$\frac{30.}{44.7} = .1079$ pound per cubic foot of steam at point on L.-P. expansion line.

$\frac{16.}{30.7} = .0758$ pound per cubic foot of steam at compression L.-P. cylinder.

$$\frac{13{,}750}{178.84} = 76.88 \qquad \frac{13{,}750}{62.31} = 220.67$$

$(.677+.08)\times.3376=.2556$ $(.238+.08)\times.1756=.0558$

$$\begin{array}{r} .2556 \\ .0558 \\ \hline .1998 \end{array}$$

$.1998 \times 76.88 = 15.36$ pounds steam at cut-off H.-P. cylinder.

$(.744+.08)\times.1079=.0889$ $(.083+.08)\times.0758=.0124$

$$\begin{array}{r} .0889 \\ .0124 \\ \hline .0765 \end{array}$$

$.0765 \times 220.67 = 16.89$ pounds steam at point on expansion line L.-P. cylinder.

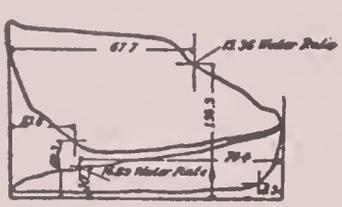


Fig. 150.

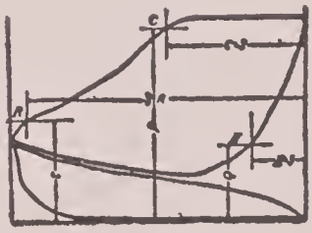


Fig. 151.

Balanced Compounds. The ideal reciprocating steam engine, stationary or locomotive, simple or compound, is an engine in which the reciprocating parts are perfectly balanced against each other, and that balancing should be accomplished without the aid of rotative counter weights. This can be done only by a correct distribution of the steam to the two or more cylinders, and then transmitting the energy developed in each cylinder, directly through the medium of its own piston rod, and con-

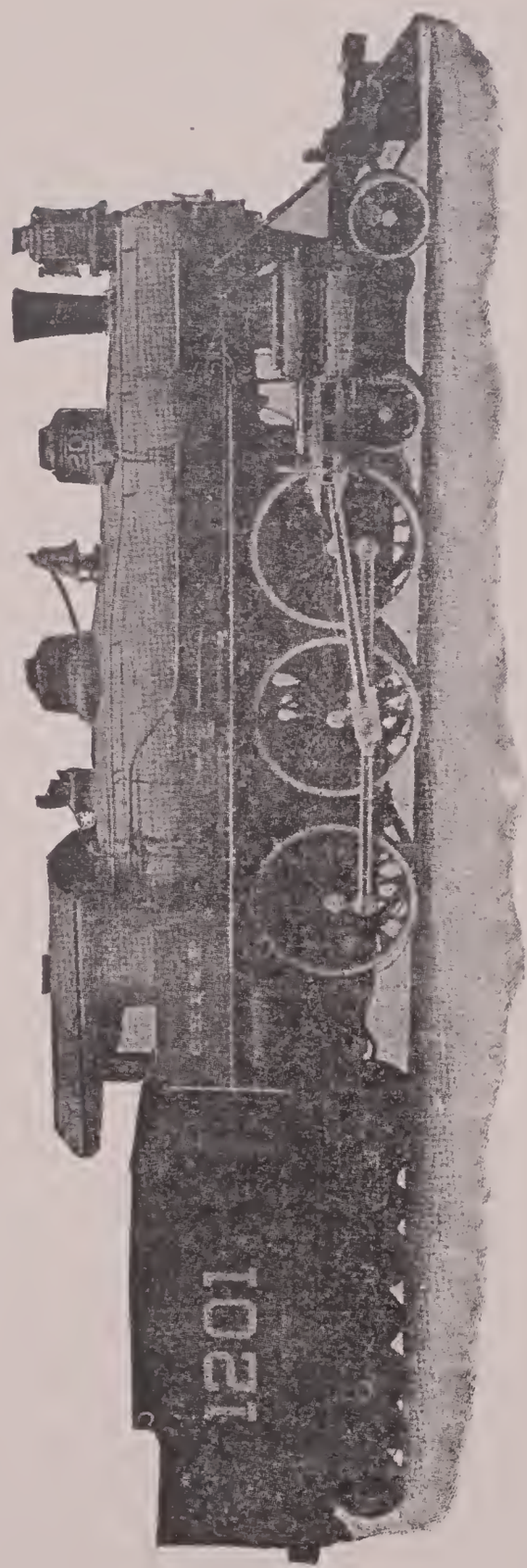


Fig. 152. Ten Wheel Passenger Engine, C. R. I. and P.

necting rod, to the engine shaft. The proper balancing of the reciprocating parts of locomotives has always been an especially serious problem, and has grown more serious with the gradual increase in the size and speed of engines. But American locomotive builders have not been timid in meeting and solving this problem, and today the four-cylinder balanced compound locomotive stands forth as a splendid specimen of mechanical ingenuity and skill in designing, and will in time, if given a square deal, prove to be the ideal locomotive.

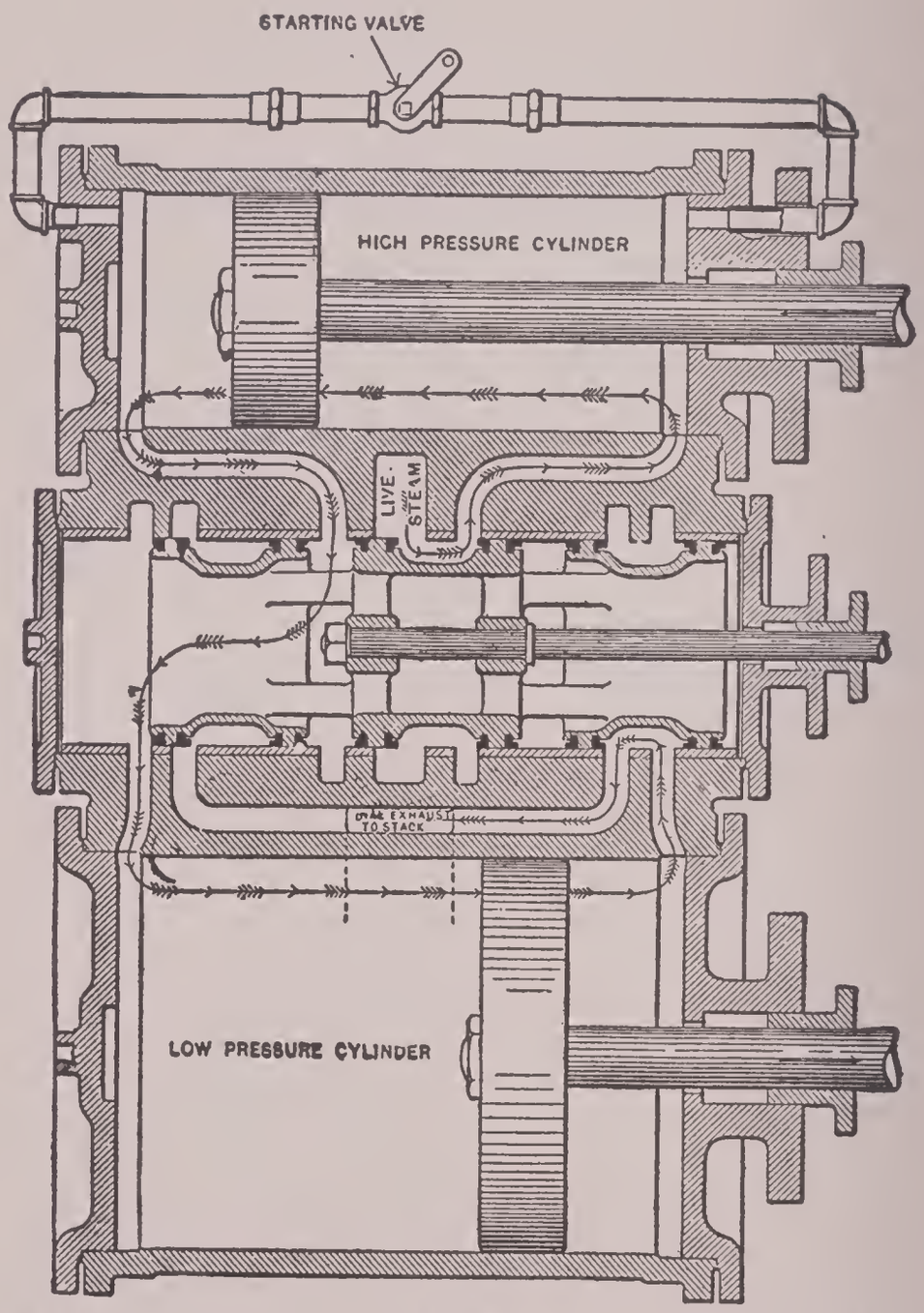
The Baldwin Locomotive Company kindly supply the following brief description of the four-cylinder balanced compound built by them:

The cylinders are a development of the original Vauclain four-cylinder compound type, with one piston slide valve common to each pair.

Instead of being superimposed, and located outside of the frames, the cylinders are placed horizontally in line with each other, the low-pressure outside, and the high-pressure inside the frames.

The slide valves are of the piston type, placed above and between the two cylinders which they are arranged to control. A separate set of guides and connections is required for each cylinder.

The two high-pressure cylinders being placed inside the frames, the pistons are necessarily coupled to a crank axle. The low-pressure pistons are coupled to crank-pins on the outside of the driving wheels. The cranks on the axle are set at 90° with each other, and at 180° with the corresponding crank-pins in the wheels. The pistons therefore travel in the opposite direction, and the reciprocating parts act against, and balance each other to the extent of their corresponding weight. The



STEAM DISTRIBUTION IN BALANCED COMPOUND CYLINDERS

Fig. 153.

distribution of steam is shown in the accompanying diagram (Fig. 153). The live steam port in this design is centrally located between the induction ports of the high-pressure cylinder. Steam enters the high-pressure

cylinder through the steam port and the central external cavity in the valve. The exhaust from the high-pressure cylinders takes place through the opposite steam port to the interior of the valve, which acts as a receiver. The outer edges of the valve control the admission of steam to the low-pressure cylinder. The steam passes from the front of the high-pressure cylinder through the valve to the front of the low-pressure cylinder, or from the back of the high-pressure, to the back of the low-pressure cylinder. The exhaust from the low-pressure cylinder takes place through external cavities under the front and back portion of the valve, which communicate with the final exhaust port. The starting valve connects the two live steam ports of the high-pressure cylinder to allow the steam to pass over the piston.

The American Locomotive Company build a four-cylinder balanced engine, having the cylinders located in practically the same manner as the Baldwin engine just described.

The use of four cylinders, two high pressure and two low-pressure, gives an opportunity for compounding under the most favorable conditions, and with each high-pressure piston working 180° from its low-pressure piston, and the other pair working 90° from the first pair, the successive impulses from the four cylinders produce a remarkably uniform turning moment. This results in a much more rapid rate of acceleration when starting up than has been possible with two-cylinder engines or with many previous types of four-cylinder engines.

The following advantages are claimed for the balanced type of locomotives, by their builders, and the claim appears to be well founded.

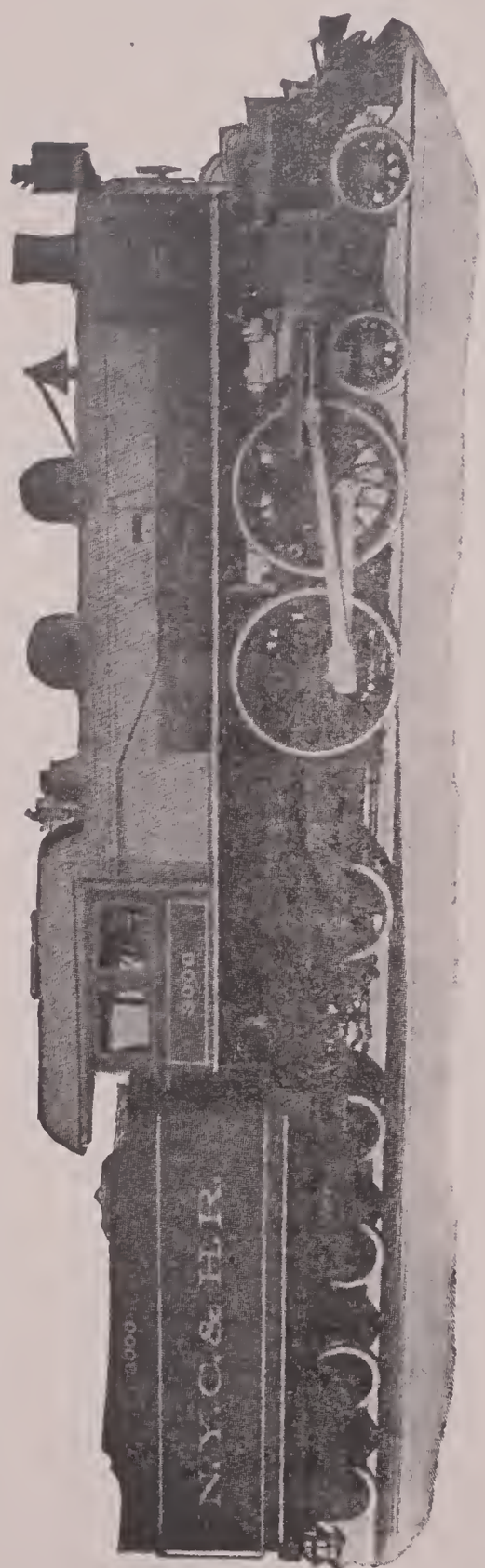


Fig. 154. 4-Cylinder Balanced Compound, Built for N. Y. C. and H. R. R.

1. The elimination of counterbalance weights from the driving wheels, the engine nevertheless being in perfect balance both horizontally and vertically. This results in the complete absence of slip at high speed.

2. The more perfect compounding which results from this arrangement of cylinders, whereby it becomes possible to secure more favorable cylinder volume ratios than with the two-cylinder compound.

3. The consequent approximately uniform turning moment throughout each revolution.

4. The power of quick acceleration, resulting partly from the uniform turning moment and partly from admitting to the low-pressure cylinders, at time of starting, and through a special starting valve, live steam at reduced pressure.

5. The reduction of stresses in the driving axles, crank-pins, and other parts of machinery, due to the system of distributing power from the cylinders, approximately one-half being transmitted to the forward driving axle, and one-half to the rear axle.

6. Increased hauling capacity and endurance at high speed, due principally to the perfection of the compounding, and the consequent economical use of steam, but partly also on account of the perfect balance of the reciprocating and revolving parts.

Tandem Compounds. Theoretically the tandem compound with its four cylinders would appear at first glance to be the ideal design, especially for locomotives, as it places the cylinders in line, and as a result of this the strains are all brought to bear along the same axis. One connecting rod, one set of guides, and one piston rod, serve to reduce the number of parts and, although there are two valves, one for the high-pressure cylinder, and

one for the low, yet one valve rod operates both valves. Notwithstanding that the tandem compound has all of these and numerous other points in its favor, it does not appear to have grown in popularity in the same degree as have the other types of compound locomotives.

One of the main objections to the tandem, and no doubt a well-founded one, is based upon the difficulties that are encountered in the examination and repair of the pistons and valves.

In many of the designs the methods that must of necessity be employed to do this work are very complicated, and consume too much time to meet with the approval of the "Boss;" and when it comes to running the engine out on the road, there are many engineers who are not studious enough, by nature, to make a success of running a compound, especially of the tandem type. A compound engine, whether marine, stationary, or locomotive, requires careful handling, more so in fact than does a simple engine, and if the engineer in charge of one expects to get good results from her, it is absolutely necessary that he should have at least an elementary knowledge of the principles upon which it is constructed, the routes of the steam passages, the construction of the valves, pistons, etc. This knowledge is easily within the grasp of every engineer, and every fireman who expects to become an engineer, and the opportunities for obtaining it are many.

The tandem compound built by the American Locomotive Company has been quite largely used in freight service during the last five years, and has met with a fair degree of success.

Cylinders. The general arrangement of cylinders and of pistons and valves is shown in Fig. 155, in which the

high-pressure cylinder is forward of the low-pressure cylinder, with both pistons on the same rod. The steam chest is common to both high and low-pressure cylinders, being open from end to end and serving the purpose of a receiver.

The valves are hollow and permit an unrestricted flow of steam through the steam chest. There being no receiver pipe on these engines, the smoke-box is fitted up with steam pipes and exhaust pipe exactly the same as in simple engines.

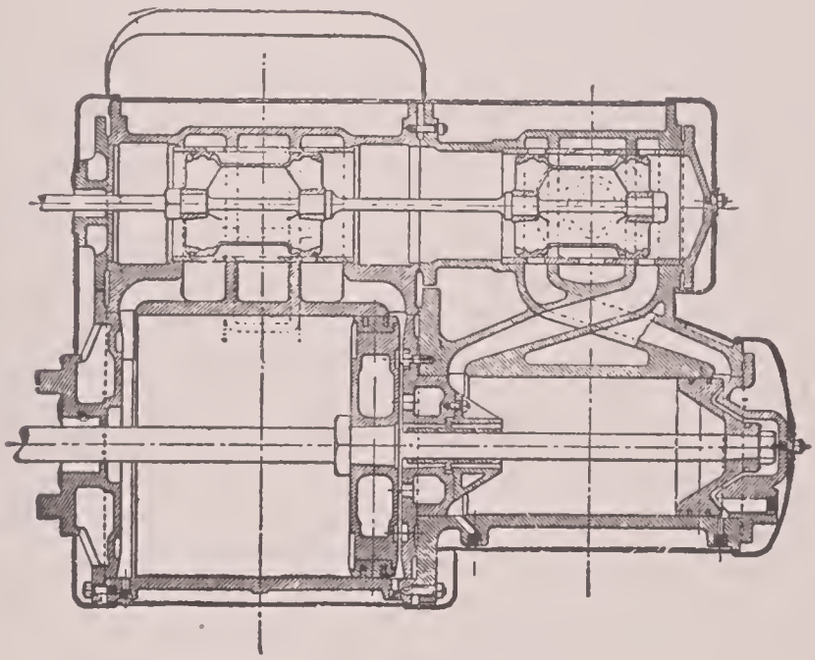


Fig. 155.

Piston Valves. On the high-pressure cylinders the valves are arranged for internal admission, and on the low-pressure cylinders for external admission. An examination of Fig. 155 will show that this design of valves allows steam to be admitted to the same side of each piston by means of the crossed ports on the high-pressure cylinder, the valves being shown as admitting steam.

Low-Pressure Cylinders. The saddle and cylinders are shown in Fig. 156 in front view and vertical section, in which the coring is shown for steam and exhaust passages. The saddle has an opening cored into the steam-pipe passage, extending from front to back on each side, where there is a circular flange for connection to the short length of steam pipe which extends from front of saddle to the high-pressure cylinder. Coring this passage through from end to end of saddle makes the cylinders interchangeable for use on either side.

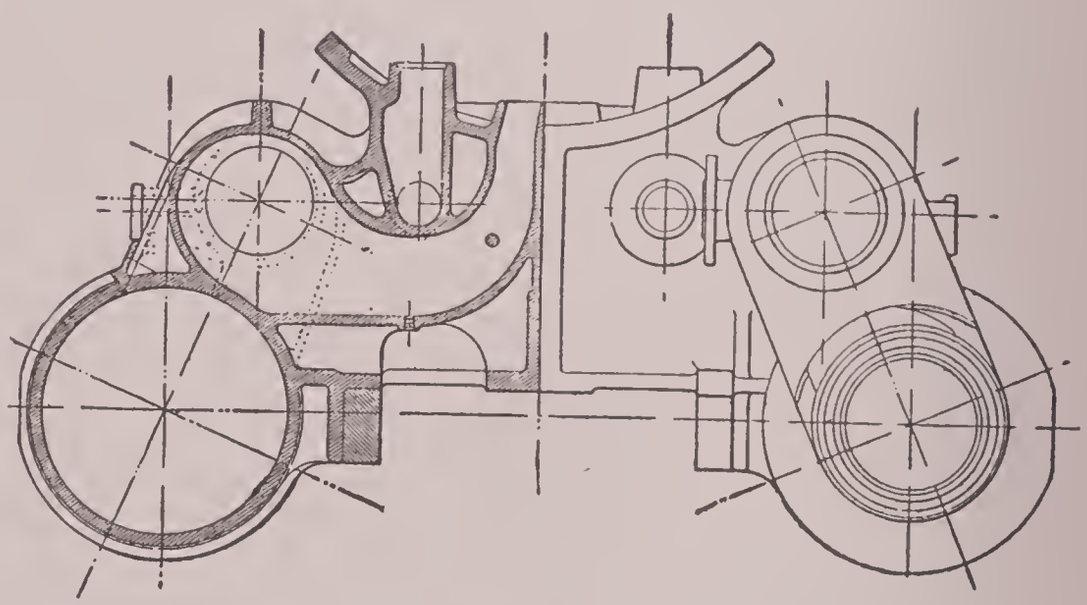


Fig. 156.

Starting Valve. To work the engine, simple or compound, at will, the starting valve shown in Fig. 157 is used, this valve being secured to the side of steam chest over the high-pressure cylinder, and having direct communication with the steam passages into that cylinder. The by-pass valves for the high-pressure cylinders are also contained in the casing of this starting valve and are worked in connection with the latter.

By-Pass Valves. For the purpose of relieving the low-pressure cylinder of excessive pressure when working steam, or freeing the same cylinder from back pressure when drifting, the by-pass valves shown in Fig. 158 are used. These by-pass valves are bolted to the side of the steam chest near each end of low-pressure cylinder, and furnish communication between the steam chest and steam ports in cylinder.

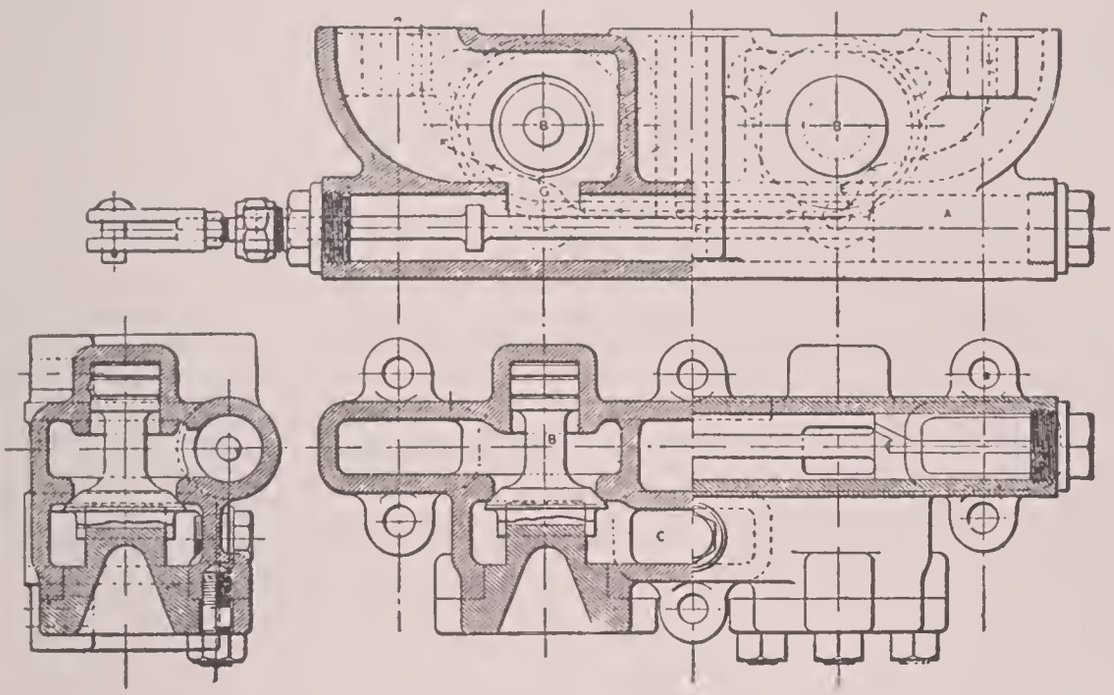


Fig. 157.

Operation, Working Simple. To start the locomotive simple—that is, to admit live steam directly to the low-pressure cylinders—the starting valve A is placed in position shown in Fig. 157 by means of a lever in the cab. Steam is admitted to high-pressure steam chest through the short steam pipe connecting saddle and chest, and passes through ports D and H, which register with the high-pressure steam ports in steam chest. From D the steam is admitted to ports E and G, and passes around the by-pass valves B, B, into port H, the valves B, B,

being held up to their seats by pressure from below through port C, which opens directly into the steam chamber of chest. Steam, having access to both high-pressure steam ports, passes through both hollow piston valves and is admitted to the low-pressure cylinder, the engine working as a simple locomotive.

Working Compound. When working compound, the starting valve A in Fig. 157 is brought to lap on port E, shutting off high-pressure steam from its passage into the low-pressure end of the steam chest. Under these conditions no steam can reach the low-pressure cylinder, except from the exhaust of the high-pressure cylinder.

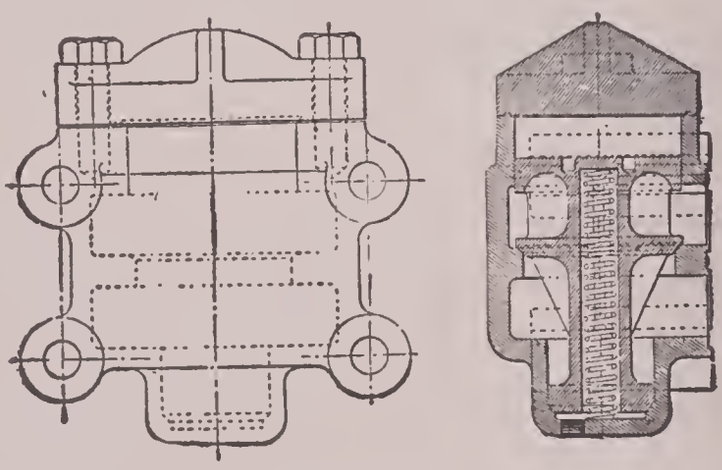


Fig. 158.

Drifting. When drifting or not working steam, the by-pass valves B, B, in Fig. 157, being in a vertical position, fall away from their seats by gravity and give a clear opening between the two ends of the high-pressure cylinder. The by-pass valves in Fig. 158 for the low-pressure cylinders are also in a vertical position, and are held to their seats by the steam chest pressure when working steam. When running with closed throttle, the by-pass valves (Fig. 158) are raised from their seats by any pressure on the lower side, assisted by the spring

under valve. With the valves raised from their seats there is a continuous opening between the two ends of low-pressure cylinder through cylinder steam ports into steam chest, providing relief from back pressure when drifting, by equalizing the pressure in the cylinders.

Starting. Any compound engine will do more economical and satisfactory work operated as a compound, and should therefore never be worked as a simple engine except in starting, or when likely to stall on grades, and then only long enough to overcome the resistance of the train.

Water. Attention should be given to the quantity of water carried in the boiler, with the view of using the steam as dry as possible. Water should not be any higher over crown sheet than is necessary for safety, since high water is not conducive to economy in operation, and is also a menace to proper lubrication.

Lubrication. When running under steam the high-pressure cylinder should receive the greater amount of oil. When drifting the reverse should be the rule, the low-pressure cylinder having the more oil.

Breakdowns. When necessary to disconnect the engine on the road, the same methods may be used as with a simple engine, as to removal of parts, blocking of cross-head, etc.

Testing Tandem Compound. The illustrations show sections through steam chests, valves and cylinders, with valves in various positions for testing. (Rules were formulated by E. P. Roesch, master mechanic, Chicago & Alton Railroad.)

It will be noticed that high-pressure valve A is central or internal admission, while low-pressure valve B is external or end admission. Also notice that ports C and

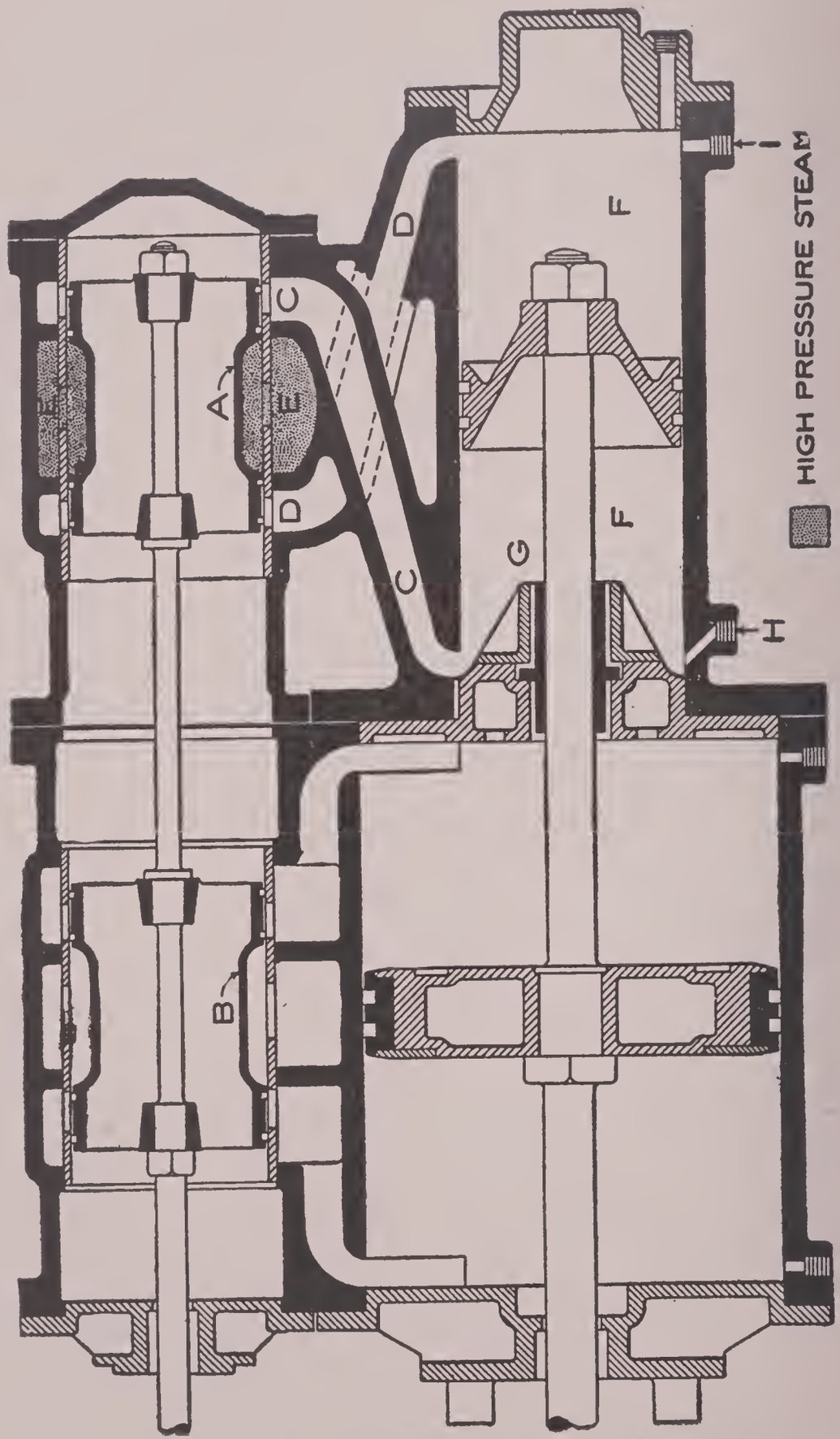


Fig. 159.

D, leading from high-pressure steam chest E to cylinder F, are crossed. Both valves A and B, and cylinder packings and piston-packing sleeve G, can be tested on each side of engine by simply moving reverse lever. To make tests, place the engine on quarter on side to be tested and proceed in manner designated on following pages.

Testing High-Pressure Valve. Engine on top quarter. Reverse lever in center of quadrant. Starting valve S closed as in Fig. 165. This places both valves A and B in central position, covering all ports on side to be tested.

By opening throttle, steam is admitted to the high-pressure steam chest E, as shown in shade. If steam now flows from either cylinder cock, H or I, Fig. 159, the high-pressure valve A is blowing.

Testing Low-Pressure Valve. Engine on top quarter. Reverse lever on center, as in Fig. 159. Starting valve S open, as in Fig. 164.

Remove by-pass valve M in Fig. 164, but replace valve-cap, which is not shown, as it is bolted to under side of starting valve. This allows steam to flow through by-pass from high-pressure steam chest E, through starting valve ports N and O, and past exhaust edges X and Y of high pressure valve A, into low-pressure steam chest P.

If steam now flows from low-pressure cylinder cocks K and L, Fig. 160, the low pressure valve B is leaking.

Testing High-Pressure Cylinder Packing. Engine on top quarter. Starting valve S closed, as in Fig. 165. Reverse lever in back motion.

This admits steam from high-pressure steam chest E, through steam port D, to front end of high-pressure cylinder F.

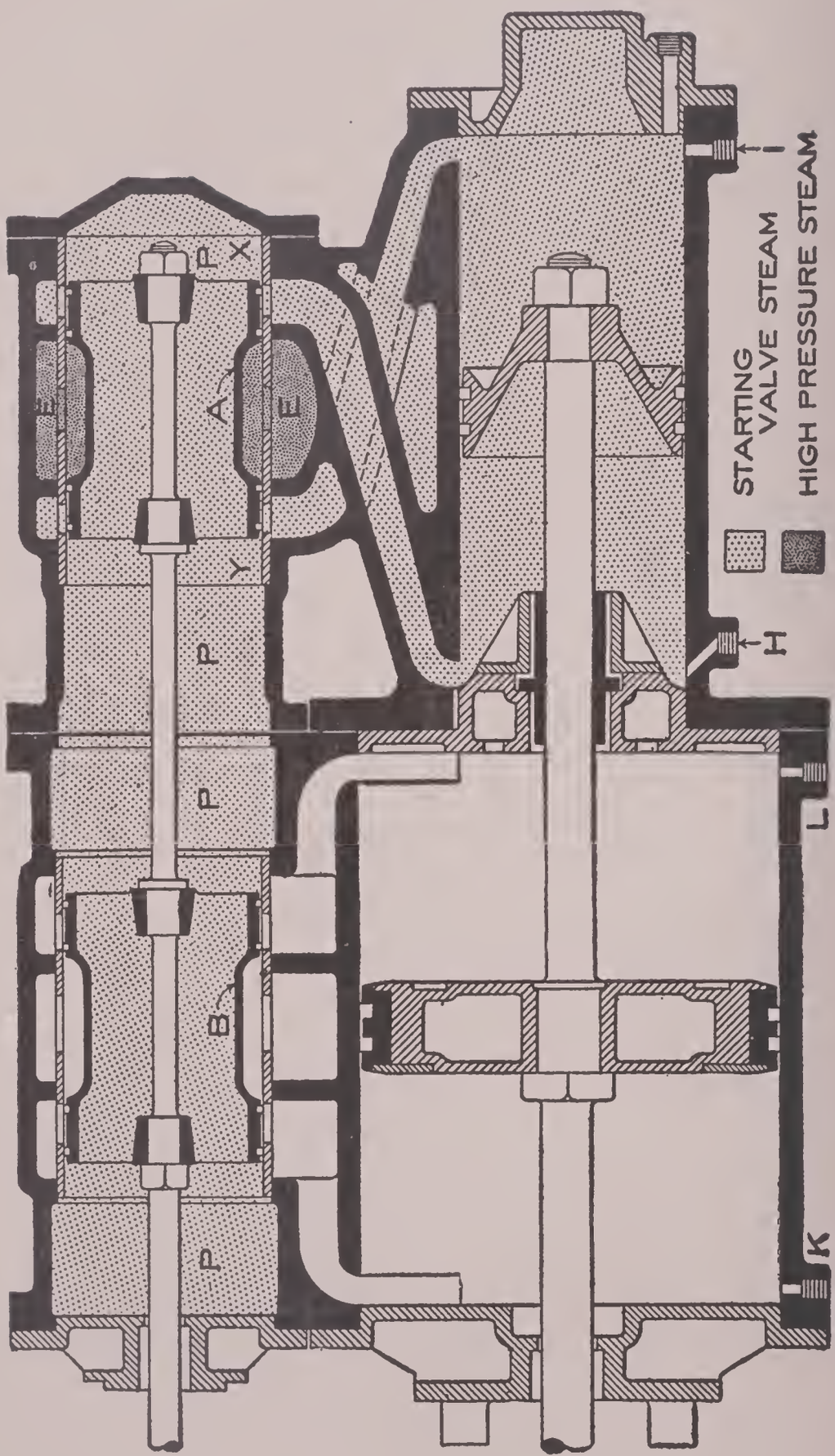


Fig. 160.

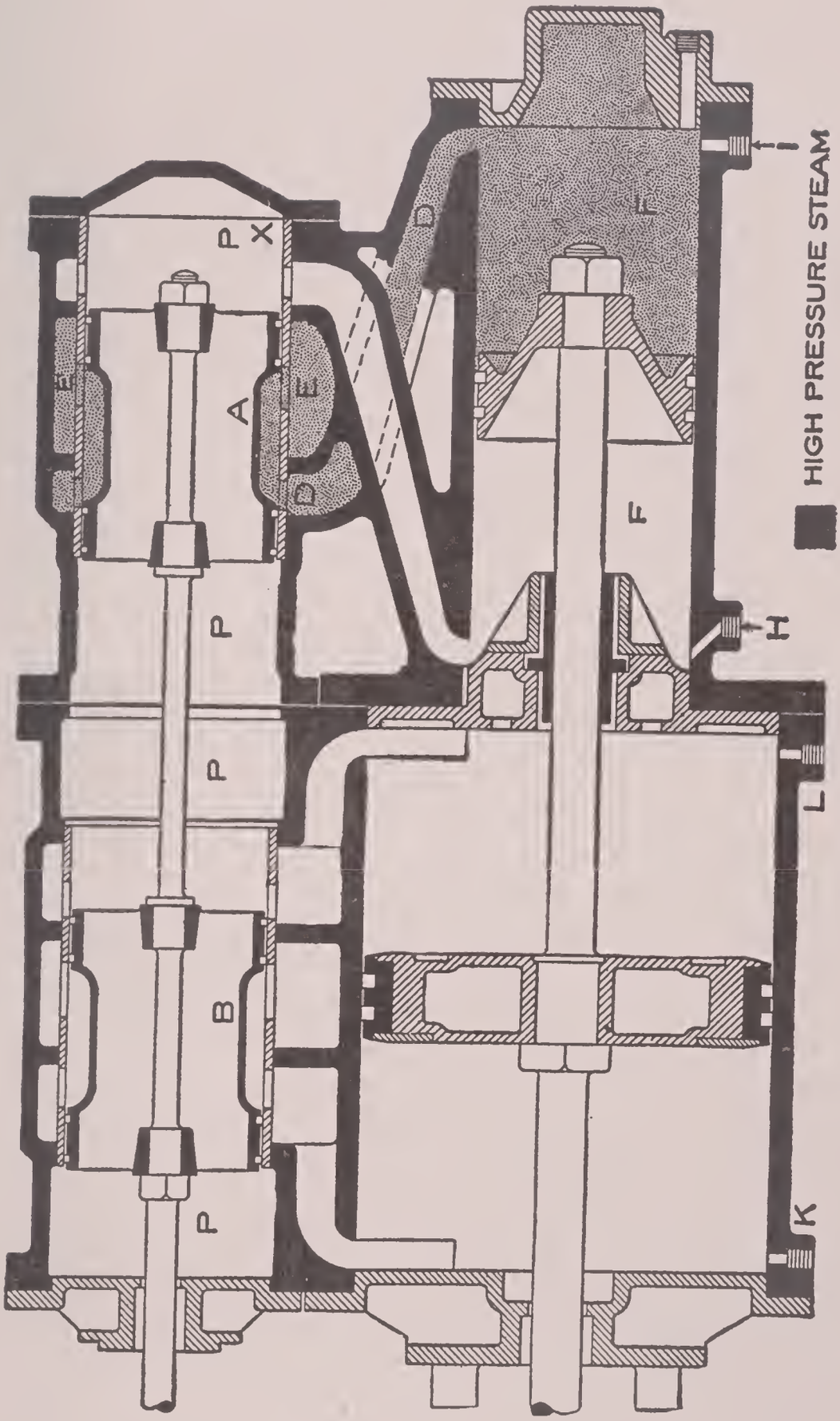


Fig. 161.

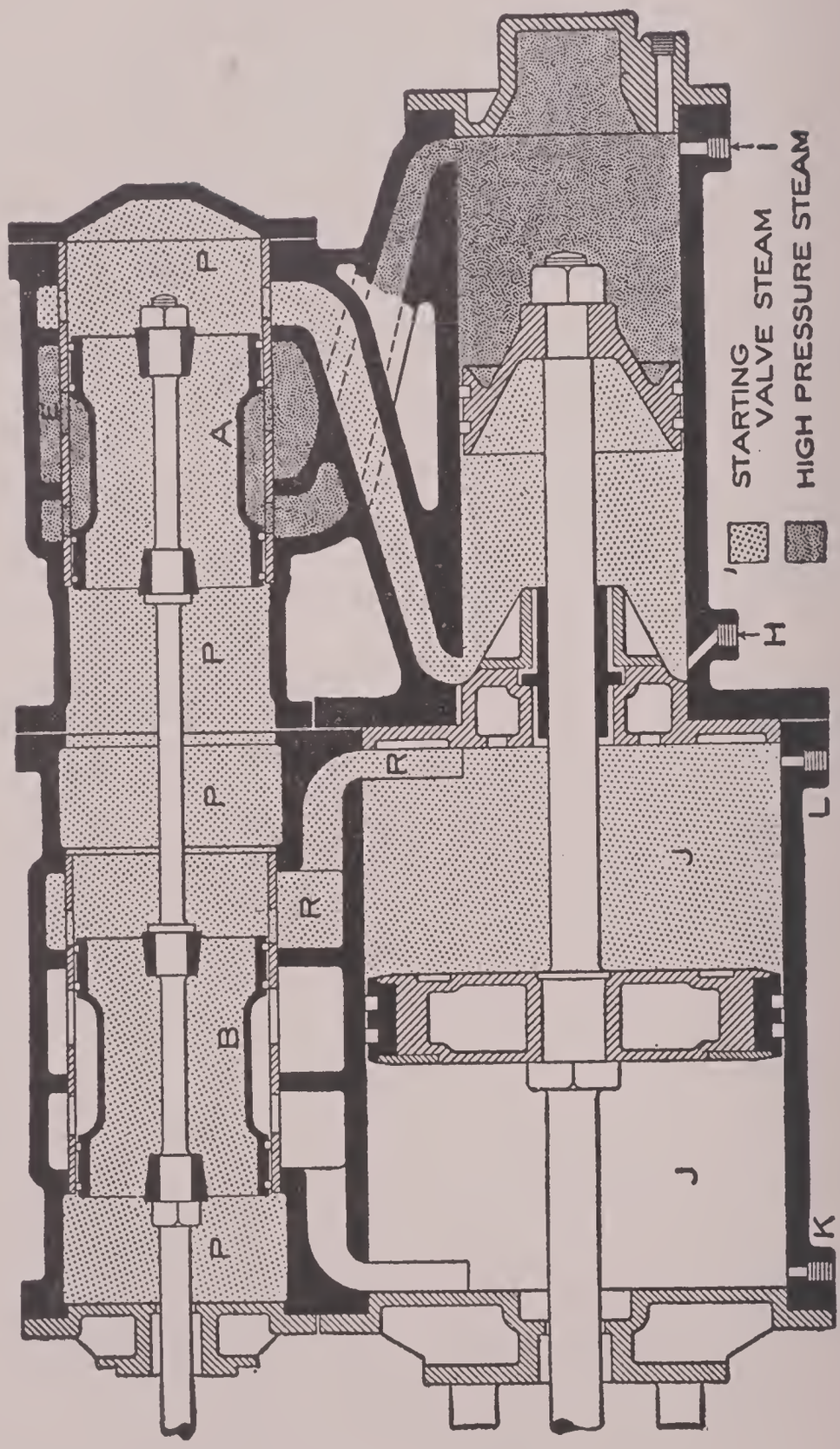


Fig. 162.

If steam now blows from back high-pressure cylinder cock H, Fig. 161, the high-pressure piston packing is blowing.

Testing Low-Pressure Cylinder Packing. Engine on top quarter. Starting valve S open, as in Fig. 164. Reverse lever in back motion. This allows steam to flow through starting valve into low-pressure steam chest P, thence through front low-pressure steam port R to front end of low-pressure cylinder J.

If any steam shows at back low-pressure cylinder cock K, Fig. 162, the low-pressure piston packing is blowing. Always test low-pressure piston packing in this position.

Testing Piston Packing Sleeve, Between Cylinders. Engine on top quarter. Starting valve S closed, as in Fig. 165. Reverse lever in forward motion. This admits steam from high-pressure steam chest E, through steam port C, to back end of high-pressure cylinder F only.

If steam now flows from front low-pressure cylinder cock L, Fig. 163, the piston sleeve G is worn and leaking.

Starting Valve in Position for Working Simple. Fig. 164 shows section through high-pressure valve, steam chest and starting valve. By-pass valve M removed, but having valve-cap replaced. For working simple, starting valve lever T should be vertical, which places valve S in forward position, opening both ports N and O.

For Fig. 160 test, the starting valve S is in position as shown in Fig. 164, but having high-pressure valve A on center, by-pass valve M removed. For Fig. 162 test, valves A and S are in position as shown in Fig. 164, but having by-pass valve M replaced.

Starting Valve in Position for Working Compound. Fig. 165, same section as Fig. 164. Both by-pass valves

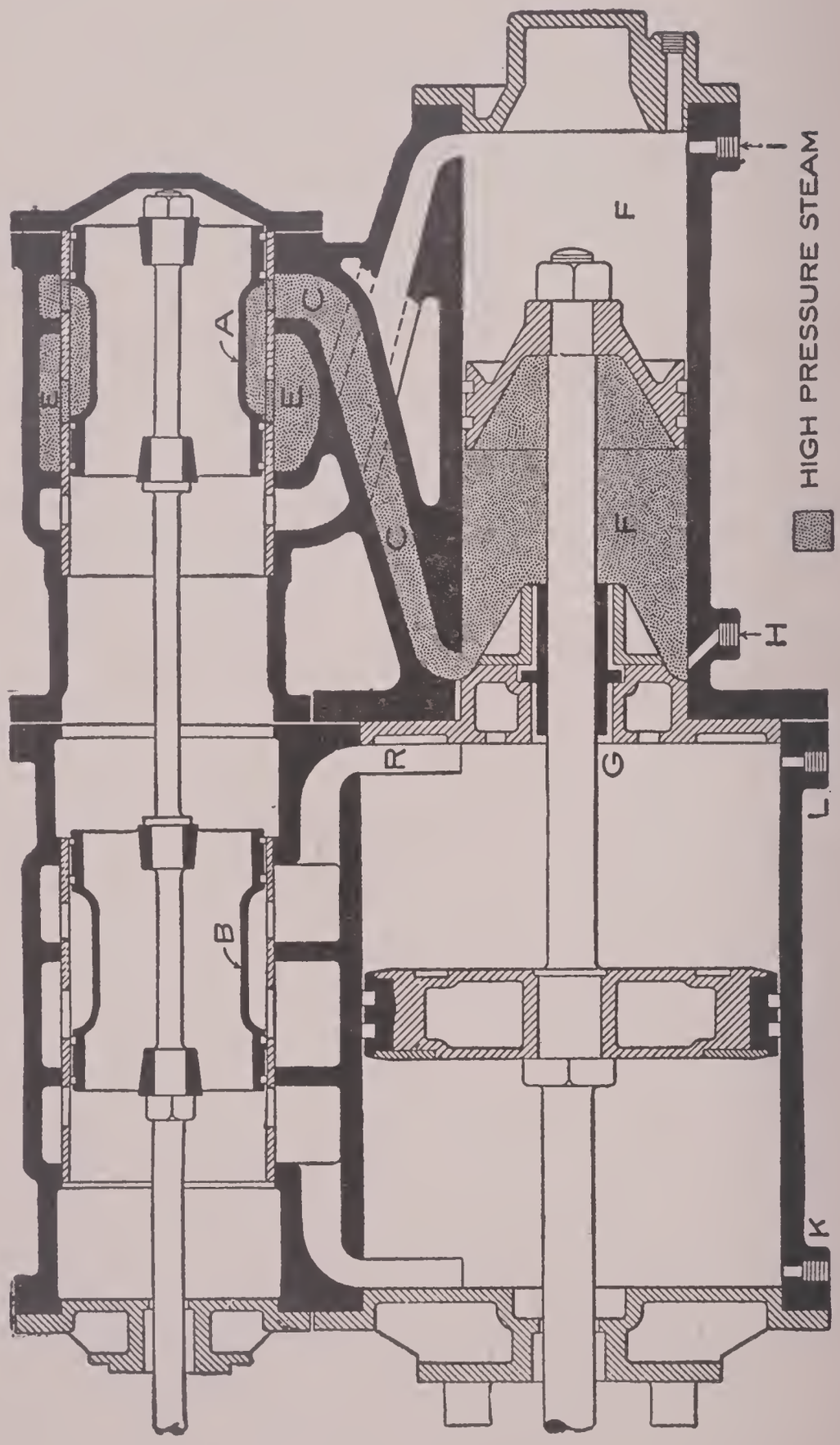


Fig. 163.

in place. Lever T in back position, so starting valve S covers port O.

For Fig. 159 test, starting valve S as in Fig. 165. The high-pressure valve A on center.

For Fig. 162 test, valves A and S in position as shown in Fig. 165.

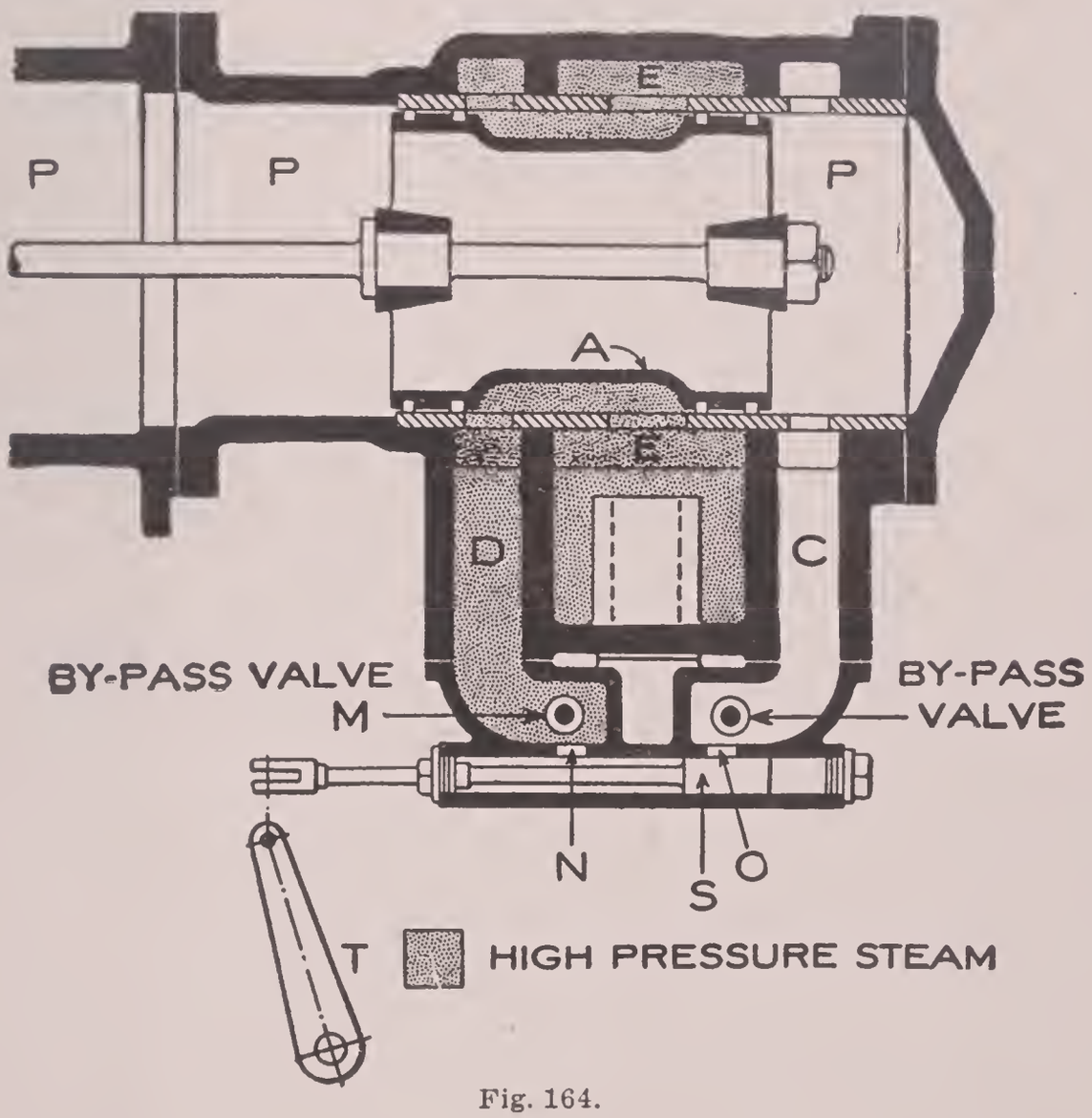


Fig. 164.

For Fig. 163 test, starting valve S as in Fig. 165. High-pressure valve A in forward motion.

The Baldwin Tandem Compound. In this type of locomotive, designed in 1902, principally for heavy freight

service, four cylinders are used, with a high and low-pressure cylinder and cylindrical valve chest on each side. The high-pressure cylinder is placed in front of the low-pressure, both having the same axis; that is, the center of the low-pressure cylinder extended becomes also the center of the high-pressure.

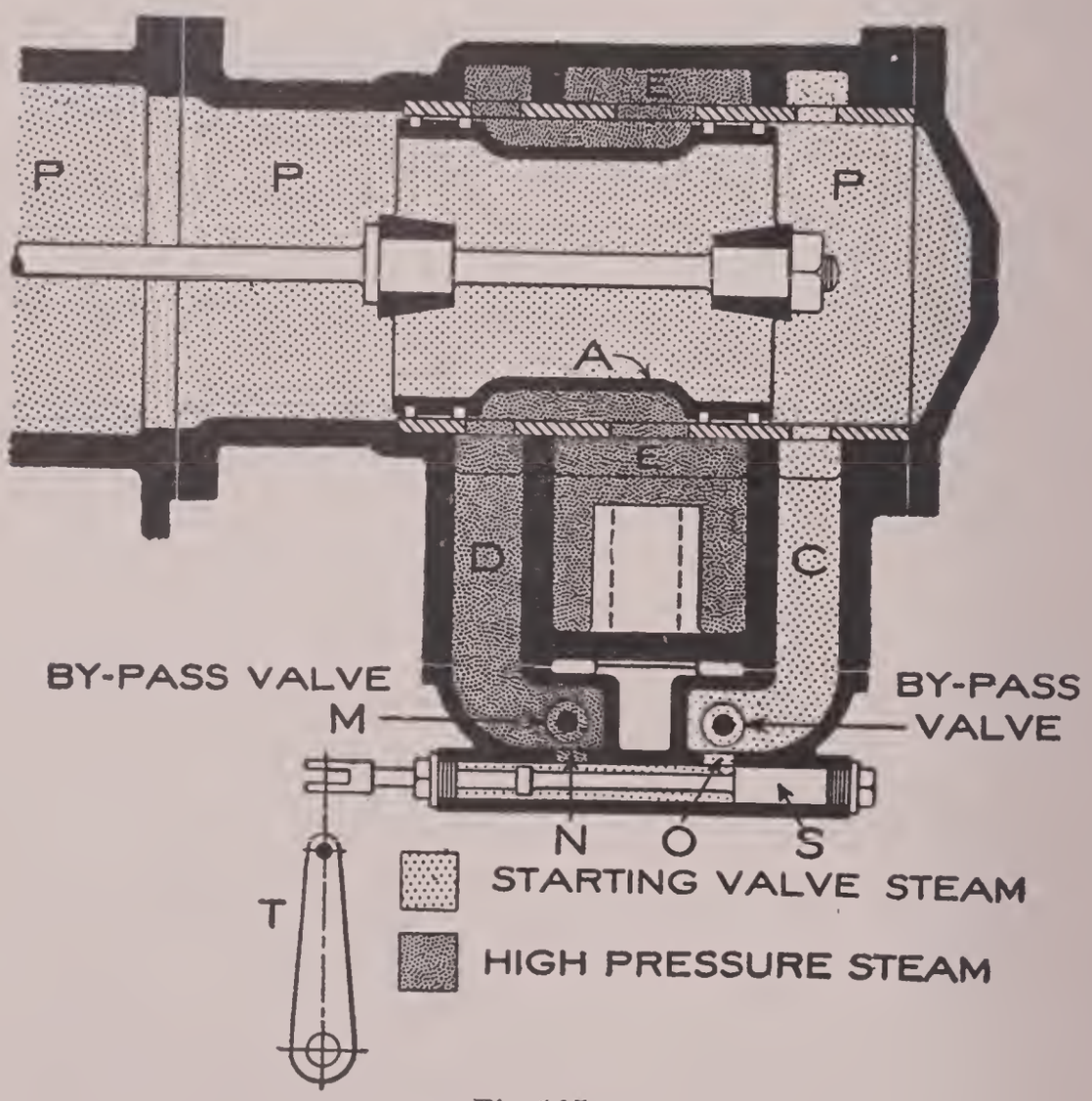
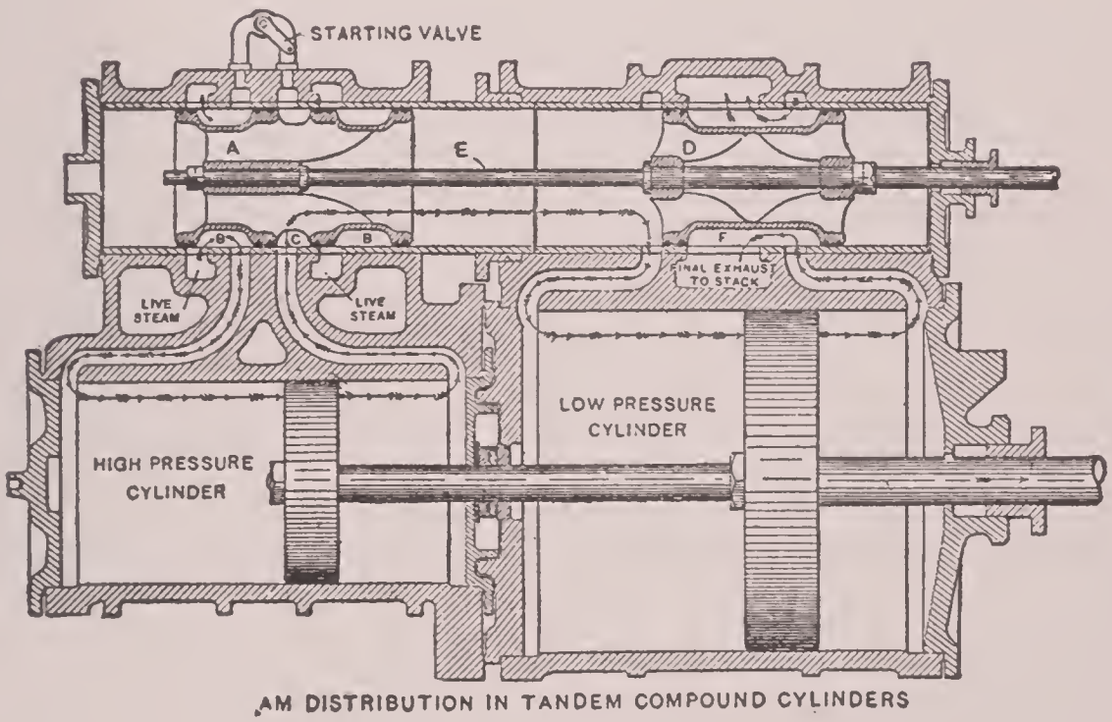


Fig. 165.

Fig. 166 is a sectional elevation of the cylinders, valve chests and valves. The arrows show the distribution of the steam.

Each cylinder with its valve chest is cast separately and is separate from the saddle. The steam connections are made by a pipe from the saddle to the high-pressure valve chest, and the final exhaust takes place through an adjustable connection between the low-pressure cylinder and the saddle casting. The valve, which is double and hollow, admits steam to the high-pressure cylinder, and at the same time distributes the high-pressure exhaust from the front end of the high-pressure cylinder to the back end of the low-pressure cylinder or vice versa, as the case may be, without the necessity of



AM DISTRIBUTION IN TANDEM COMPOUND CYLINDERS

Fig. 166.

crossed ports. As shown in the accompanying diagram, Fig. 166, A is the high-pressure valve by which steam is conducted from the live-steam openings through external cavities B and B to the high-pressure cylinder. The exhaust from the high-pressure cylinder passes through the opening C to the steam chest, which acts as a receiver;

D is the low-pressure valve connected to the high-pressure valve by valve rod E. This valve in its operation is similar to the ordinary slide valve. The outside edges control the admission, and the exhaust takes place through the external cavity F. The starting valve connects the live-steam ports of the high-pressure cylinder.

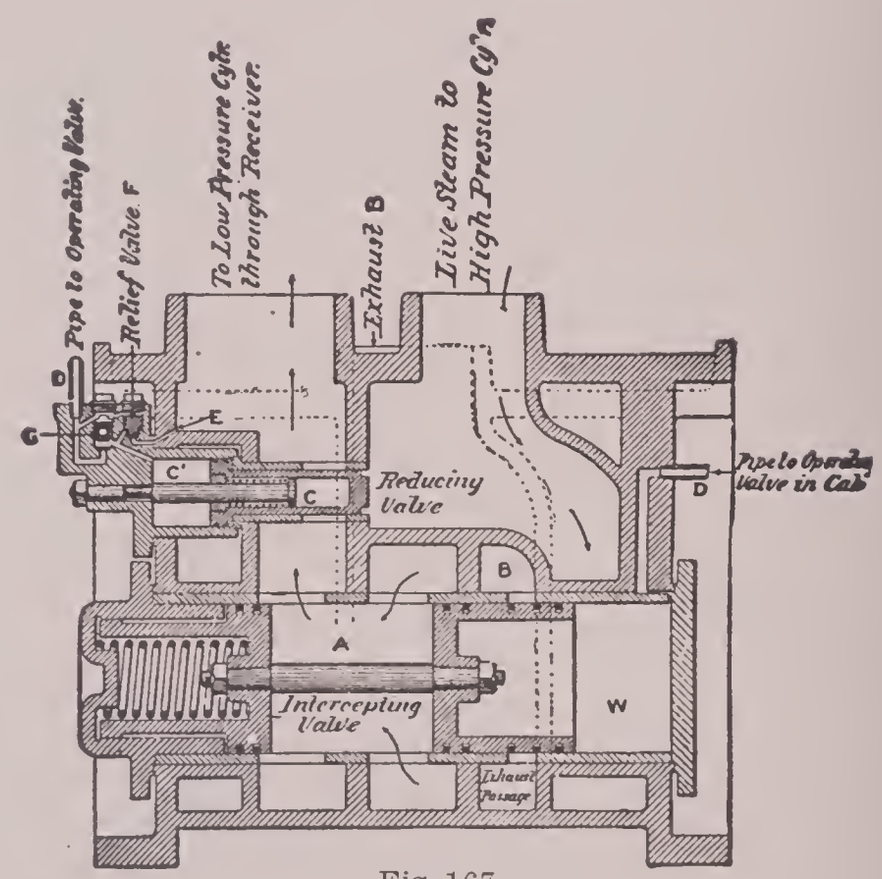


Fig. 167.

The Cross Compound. The cross compound locomotive has two cylinders, one on each side, with an intercepting valve, so arranged that the engineer can work the engine either simple or compound. When the engine is worked as a simple engine, the pressure of the steam that is admitted to the low-pressure cylinder is controlled by an automatic reducing valve in such a manner that it shall bear the same ratio to the pressure of steam admitted to the high-pressure cylinder as the volume of

the high-pressure cylinder bears to the volume of the low-pressure cylinder. Unequal strains are thus avoided. As previously stated, a compound locomotive should never be worked as a simple engine, except in starting a heavy train, or when there is danger of getting "stuck" on a heavy up grade.

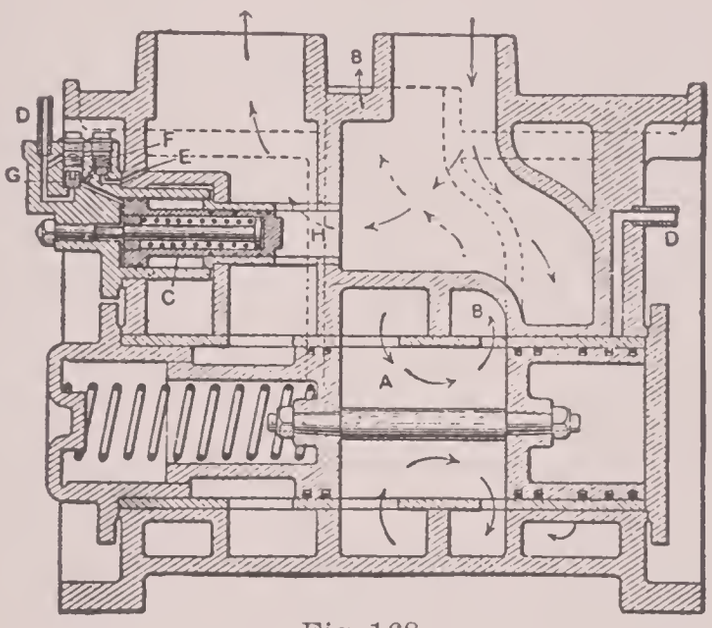


Fig. 168.

The Baldwin Two-Cylinder Compound. The essential features of this design, brought out in 1898, are the intercepting and the reducing mechanisms. These, when in normal position, permit the locomotive to operate by single expansion, and so continue until changed to compound. The engine is therefore readily started at any position of the crank.

In the diagrams, Figs. 167 and 168, A is a double piston intercepting valve, located in the saddle casting of the high-pressure cylinder. In one direction the movement is controlled by a spiral spring, in the other by steam pressure. The function of the intercepting valve is to cause the exhaust from the high-pressure

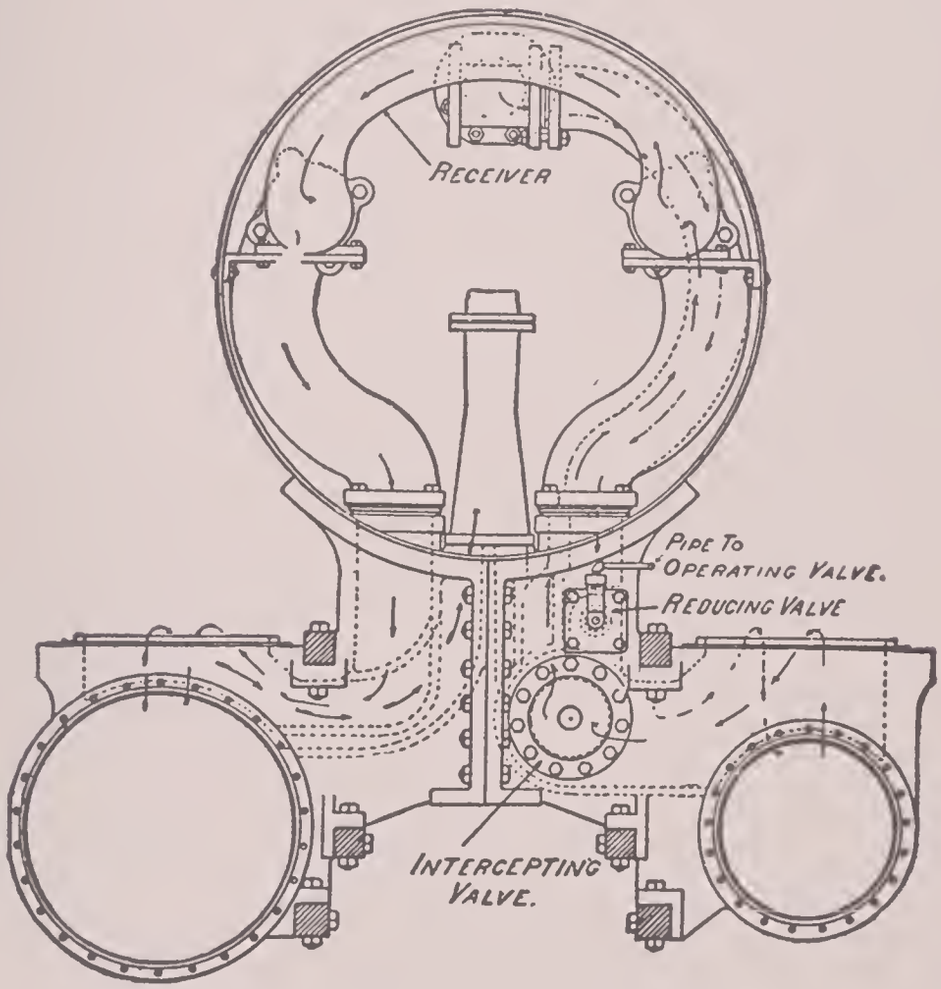
cylinder to be diverted, at the option of the engineer, either to the open air when working single expansion, or to the receiver when working compound. C is a reducing valve, also placed in the saddle casting of the high-pressure cylinder, and like the intercepting valve is moved in one direction by a spiral spring, and in the opposite direction by steam pressure. The function of this valve is, in its normal position, to admit live steam into the receiver at reduced pressure while the locomotive is working single expansion. When the engine is working compound, this valve automatically closes, as it is evident that there is no further need of live steam in the receiver.

A further function of the reducing valve is to regulate the pressure in the receiver so that the total pressure on the pistons of the high and low-pressure cylinders may be equalized.

The steam for controlling the operation of both intercepting and reducing valves is supplied through pipes D from the operating valves in the cab. When not permanently closed by pressure in the pipes D, the reducing valve C is operated automatically by the pressure in the receiver. To this end the port E is provided, communicating with the receiver, and the space in front of the reducing valve; as the pressure rises the steam acts on the large end of the reducing valve, causing it to move backward and close the passage H, through which steam enters the receiver, and thus prevent an excess pressure of steam in the low pressure cylinder.

Poppet valves F and G are placed in connection with port E, one to prevent the escape of steam from the receiver to pipe D when the locomotive is working single expansion, and the other to close the passage from pipe D to the receiver when working compound. Normally

the lever of the operating valve in the cab is in the position marked "simple." In this position no steam is allowed to enter the pipes D, and no pressure will be exerted on the intercepting and reducing valves in opposition to the springs, and they will assume the positions shown in Fig. 167.



TWO-CYLINDER COMPOUND. CROSS-SECTION

Fig. 169.

The ports of the intercepting valve A stand open to receive the exhaust steam from the high pressure cylinder, and deliver it through the exhaust passage B to the atmosphere.

The reducing valve is open, admitting live steam through passage H to the receiver, and from thence to the low-pressure cylinder.

The receiver pressure is governed by the automatic action of the reducing valve, as previously explained. In this way the engine can be used single expansion in making up and starting trains, for switching and slow running.

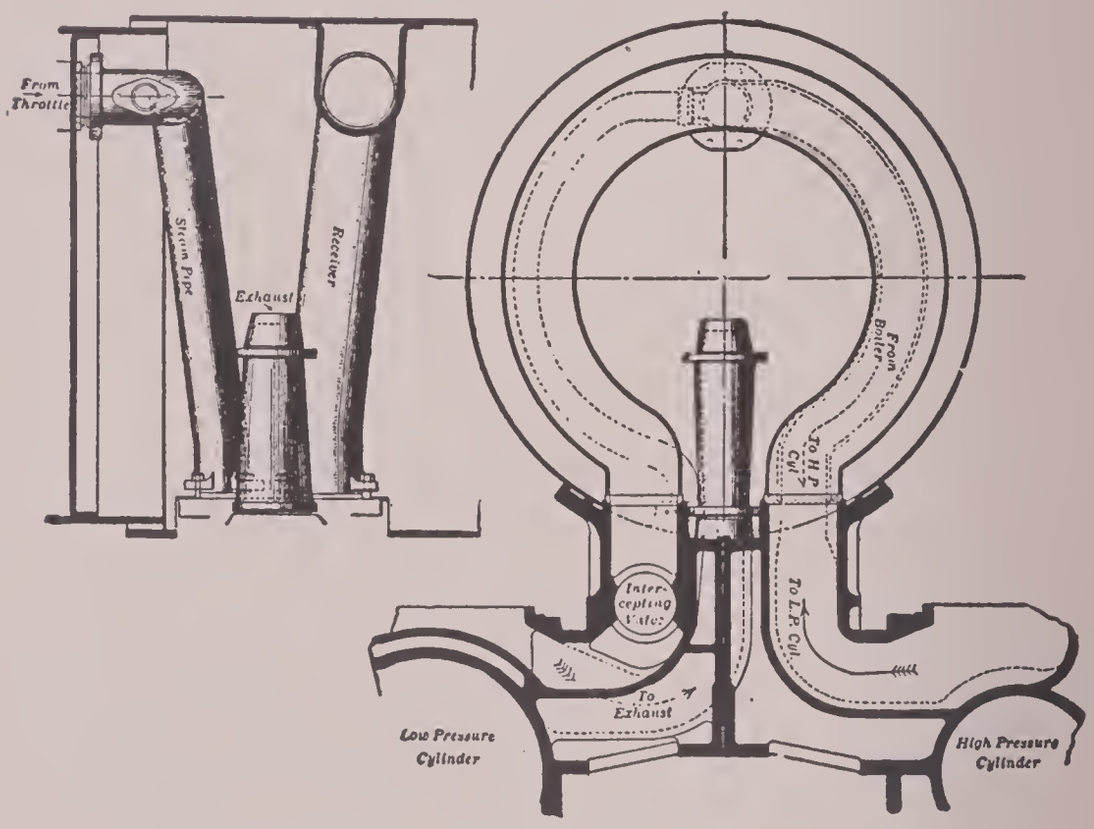


Fig. 170.

At the will of the engineer the operating valve in the cab is moved to the position marked "compound." This admits steam to the pipes D, and through them to the valve chambers W and C^1, changing the intercepting and reducing valves instantly and noiselessly to the pistons shown in Fig. 168. The exhaust from the high-pressure cylinder is diverted to the receiver,

the admission of live steam to the receiver is stopped by the closing of the passage H, and the locomotive is in position to work compound.

Both valves are of the piston type, with packing rings to prevent leakage. This insures an easy movement of the valves, and prevents the hammering action common to valves of the poppet type when automatically operated.

Schenectady Cross Compound. The American Locomotive Company kindly furnish the following description of the two-cylinder cross compound engine as built at their Schenectady works.

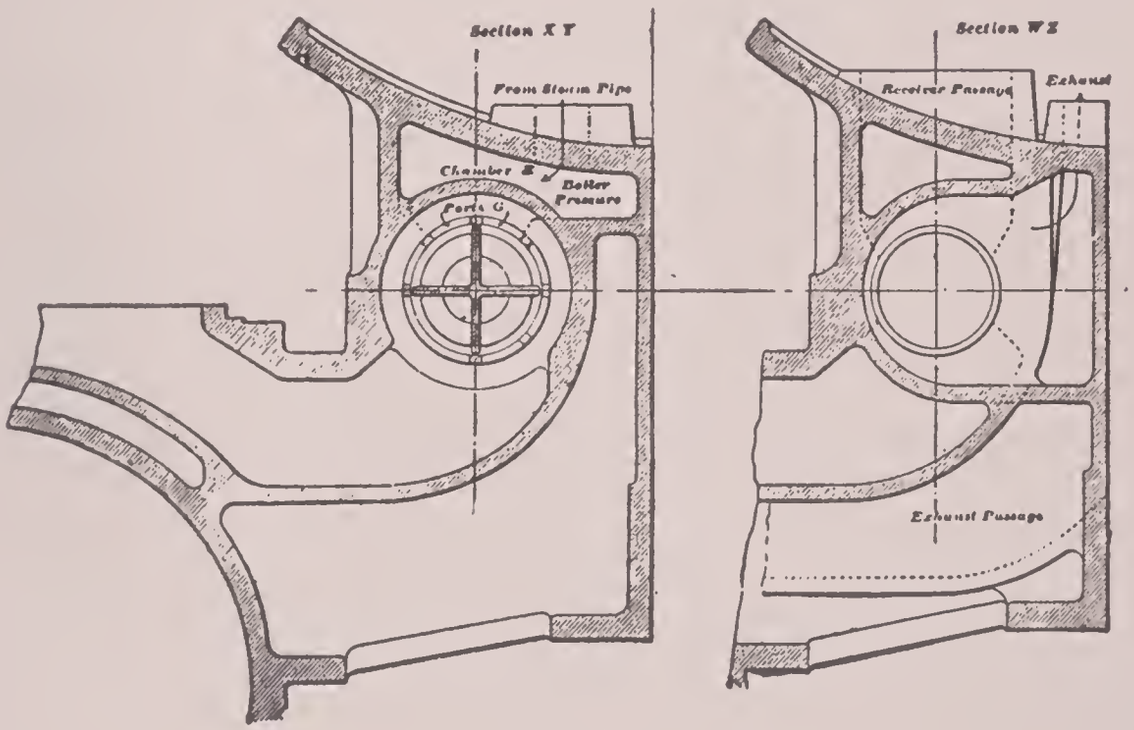


Fig. 171.

Figure 170. A sectional view through the smoke arch and cylinder saddles, showing the steam passages, receiver, and the location of the intercepting valve in the low-pressure cylinder saddle.

Fig. 171. A transverse section through the low-pressure cylinder saddle XY and WZ. Section XY shows

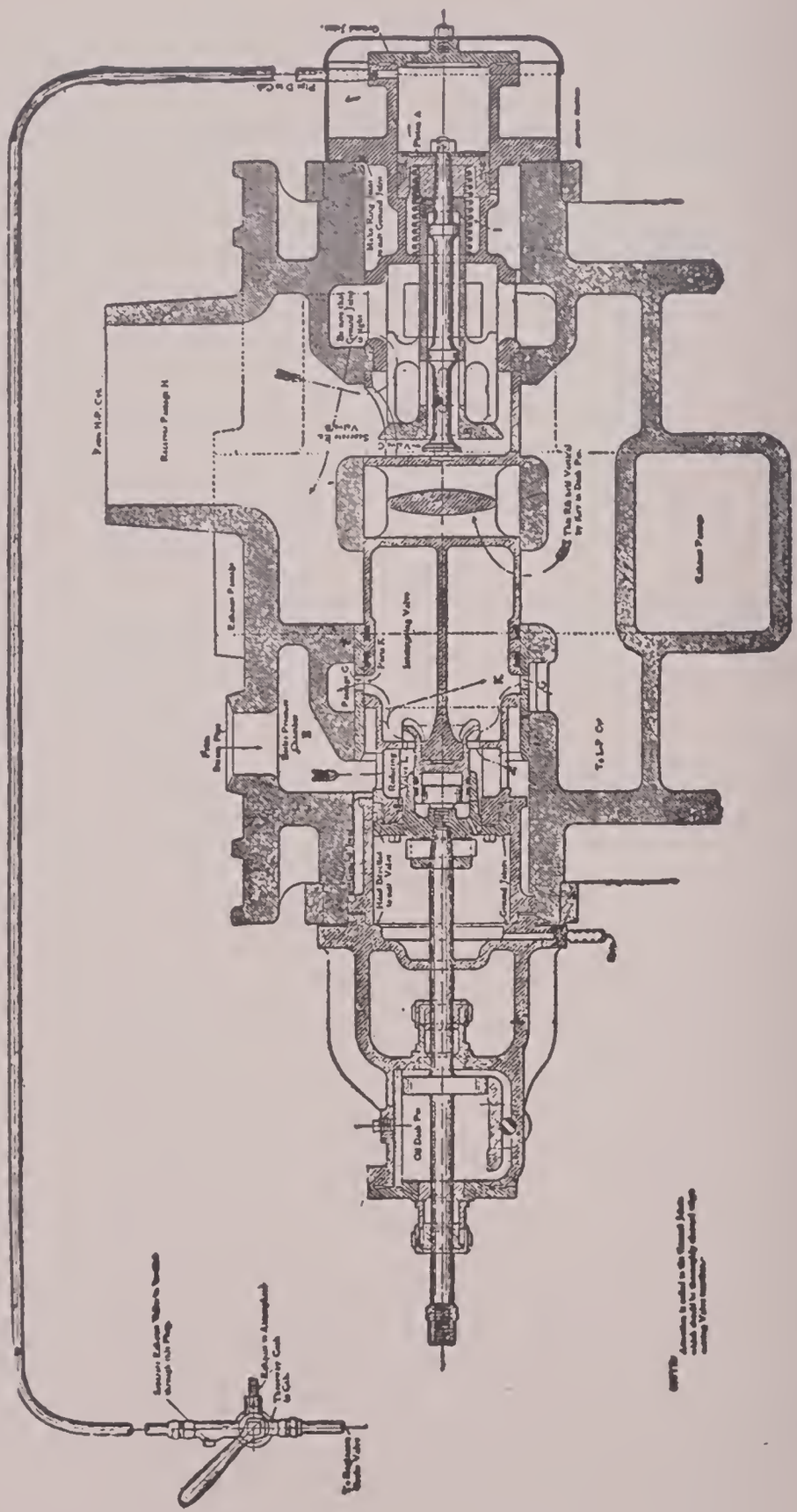


Fig. 172.

the passages for admitting live steam into the low pressure cylinder, and section WZ the outlet passage from separate exhaust valve to the exhaust pipe.

Fig. 172. A vertical section through the low-pressure cylinder saddle and intercepting valve, showing the intercepting and separate exhaust valves in the position taken when engine is working simple.

Fig. 173. The same section as Fig. 172, but shows the position of the intercepting and separate exhaust valves, when the engine is working compound. With the arrangement of valves shown in these figures the engine can be started and run either compound or simple, and can be changed from compound to simple, or from simple to compound, at the will of the engineer.

General Description. As the throttle is opened, steam from the boiler, through the dry pipe, is admitted directly to the high-pressure steam chest, and at the same time to chamber E, surrounding the reducing valve L, Figs. 172 and 173.

The exhaust from the high-pressure cylinder, by means of the receiver pipe, passes to chamber surrounding the intercepting valve, and thence to the low-pressure steam chest when working compound, intercepting valve in position in Fig. 173, or to the atmosphere, through separate exhaust valve and stack, when working simple, valve in position shown in Fig. 172.

The low-pressure exhaust passes directly to the stack at all times.

The intercepting valve opens and closes the connection between the two cylinders.

The separate exhaust valve opens and closes the connection between the high-pressure cylinder and the atmosphere.

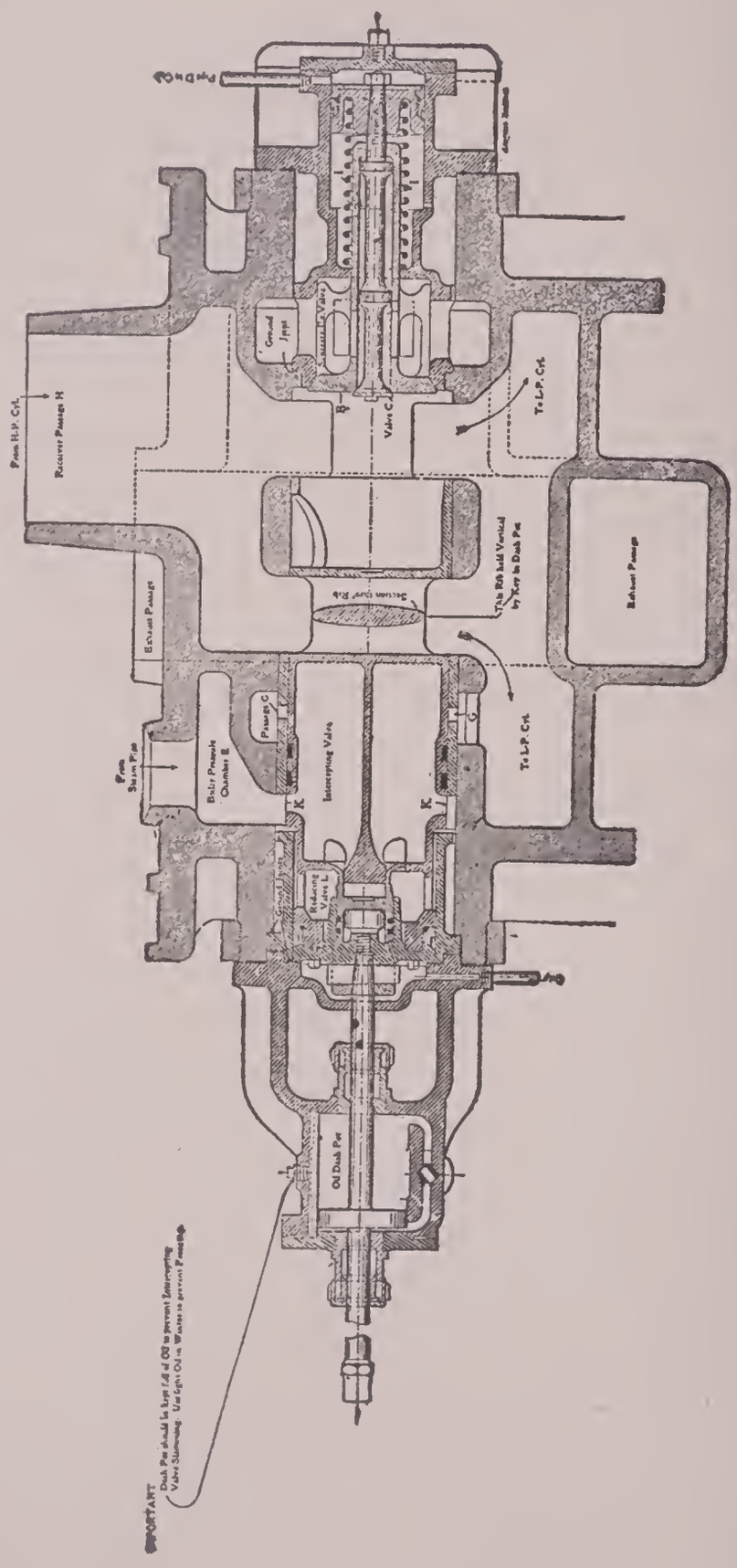

From H.P. Cyl.
Receiver Passage H
Ground Joint
Valve Chest
To L.P. Cyl.
Exhaust Passage
This Rib held Vertical by Key in Dash Pot
Exhaust Passage
From Steam Pipe
Boiler Pressure Chamber E
Passage C
Intercepting Valve
To L.P. Cyl.
K
K
Reducing Valve L
Oil Dash Pot
IMPORTANT
Dash Pot should be kept Full of Oil to prevent Intercepting Valve Slamming. Use light Oil to Water to prevent Passage

Fig. 173.

The function of the reducing valve, which operates only when the engine is working simple, or starting, is to control the admission of steam from the boiler to the low-pressure cylinder, in order that the pressure of steam admitted to the low-pressure cylinder shall have the same ratio to the steam in the high-pressure cylinder as the volume of the high-pressure cylinder is to the volume of the low-pressure cylinder.

The oil dash pot insures a steady movement of the intercepting valve.

The intercepting and reducing valves operate automatically by means of the steam pressure acting on the difference of areas of the ends of the valves. The movement of the reducing valve is cushioned by the small air dash pots shown. The separate exhaust valve is operated by the engineer, by means of a three-way cock in the cab. To open the separate exhaust valve, the handle of the three-way cock is moved to the piston provided for admitting pressure against the piston A, Fig. 172. Moving the handle in the opposite direction relieves the pressure against A, and the spring, which is shown in the figure, shuts the valve. The separate exhaust valve can be so connected as to operate either by air or steam.

Operation, Starting Simple. The handle of the three-way cock in the cab is moved by the engineer so as to admit pressure through the pipe D against the piston A, forcing it and the valves B and C to the position shown in Fig. 172. As the throttle is opened, steam is admitted directly from the boiler into the passage E, forcing the intercepting valve into the position shown (Fig. 172), thence the steam passes through the intercepting valve by the ports K K, and the passage G G, through the re-

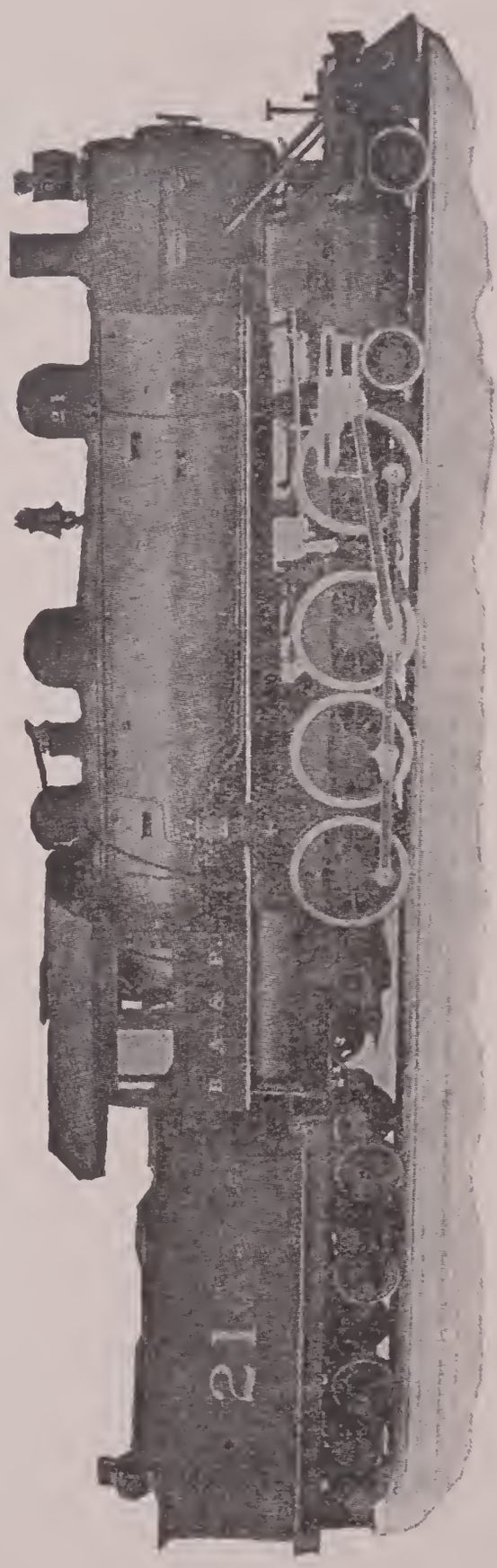


Fig. 174. Schenectady 2-Cylinder Compound, Built for the Butte, Anaconda and Pacific Railway.

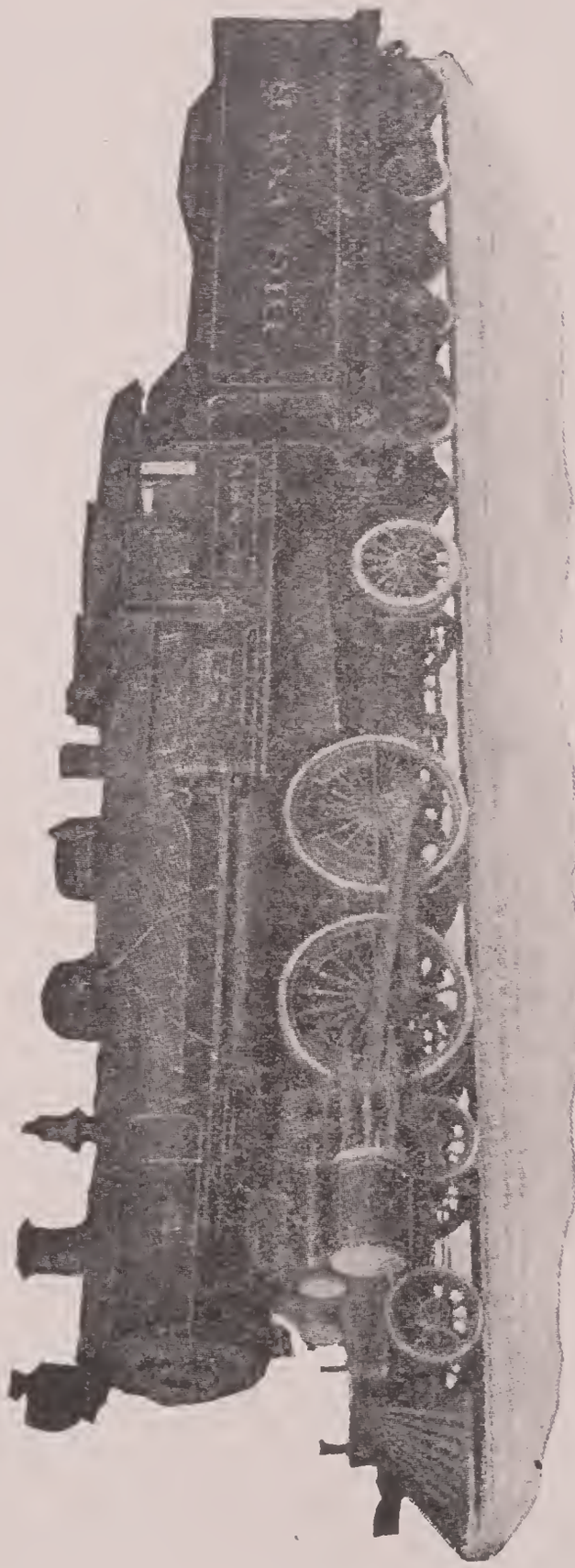


Fig. 175. Passenger Locomotive, Built for Cleveland, Cincinnati, Chicago and St. Louis Railway.

ducing valve to the low-pressure steam chest; at the same time steam from the boiler is admitted directly, by means of the steam pipe, to the high-pressure steam chest. The exhaust from the high-pressure cylinder passes to the atmosphere by means of the receiver passage H and the separate exhaust valve B. Steam from the low-pressure cylinder is exhausted directly to the atmosphere.

To Change from Simple to Compound. Having started simple, to change to compound, the handle of the three-way cock in cab is turned so that pressure is released from the piston A. The separate exhaust valve will then be closed by the spring I. The pressure in the receiver, due to the exhaust from the high-pressure cylinder, will rise and force the intercepting valve to the left, that is, to the position shown in Fig. 173, thereby opening the passage for the exhaust steam, from the high-pressure cylinder, through the receiver, to low-pressure steam chest. The movement of the intercepting valve to the left also closes the passage G G, thereby shutting off the admission of steam directly from the boiler to the low-pressure steam chest.

Starting Compound. To start the engine compound the separate exhaust valve is left closed as in Fig. 173. As the throttle is opened the steam pressure in the passage E will force the intercepting valve to the right or to the closed position; at the same time steam directly from the boiler will be admitted to low-pressure steam chest through ports K K and passage G G. The high-pressure cylinder will exhaust into the receiver until the pressure is sufficient to force the intercepting valve to the left, as shown in Fig. 173, when the engine will work compound. The change to compound working

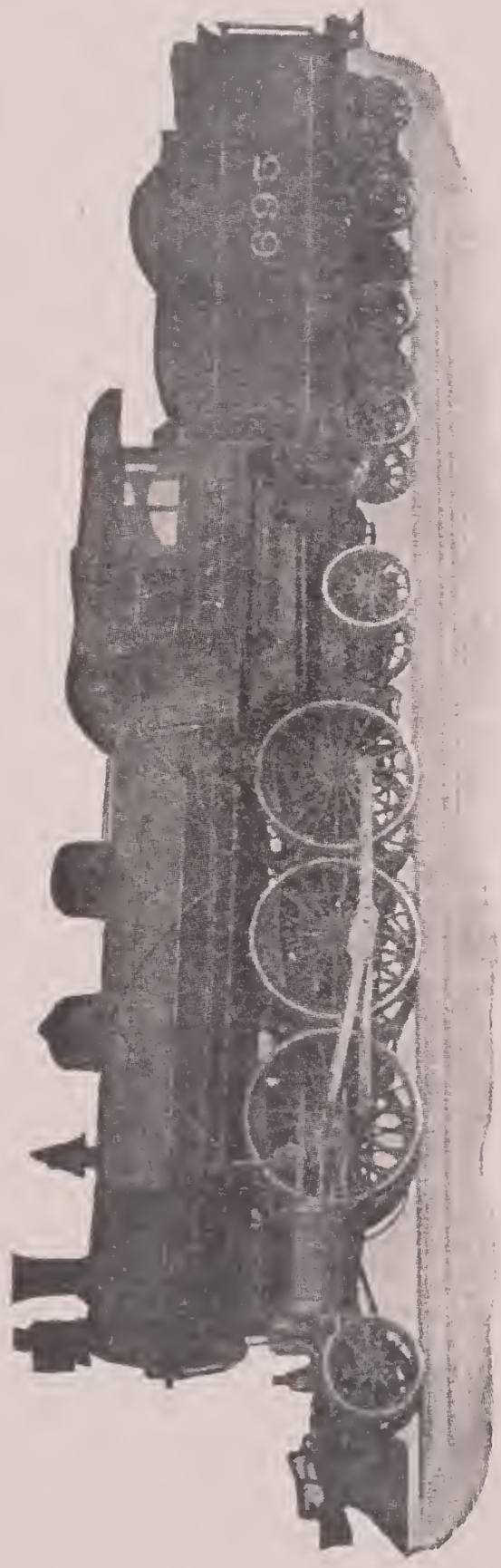


Fig. 176. Passenger Locomotive, Built for Lake Shore and Michigan Southern Railway.

takes place at from one-half to three-quarters of a revolution of the driving wheels.

Compound to Simple. With the engine working compound, if the engineer wishes to run the engine simple to prevent stalling on a heavy grade, the handle of the three-way cock should be placed in same position as for starting simple. This opens first the small bleeding valve C, Figs. 172 and 173, and then the separate exhaust valve. The bleeding valve relieves the pressure and thus permits the main valve B to be operated more easily. As soon as the separate exhaust valve is open, the pressure in the receiver drops and the intercepting valve is forced against the seat to the right, by means of the pressure in chamber E, and the engine works simple as before. Engines should be worked simple no longer than absolutely necessary.

Lubrication. A pipe from the sight feed lubricator located in the cab leading directly to chamber E is provided, by means of which both the intercepting and reducing valves are lubricated. One drop per minute is sufficient for these parts. A small oil cock in three-way cock, located in cab, provides for lubricating the separate exhaust valve and attendant parts, and oiling once a day with a small quantity of cylinder oil provides sufficient lubrication.

When using steam it is good practice to feed about two-thirds of allowance of cylinder lubrication to H.-P. cylinder. When drifting down long grades this should be reversed, on account of the larger surface to be lubricated on L.-P. side. Always run with lubricator steam valve wide open.

By-Pass Valves. Some of the compound locomotives recently built are equipped with by-pass valves, pro-

vided to admit of engines drifting more freely. These valves, more particularly on the low-pressure side, should be examined occasionally, by removing the cap, to insure that they are in good working order, On new engines the by-pass valves should be cleaned frequently, as their free movement is liable to be hindered by gumming or the presence of core sand.

Should a by-pass valve become broken or in any way defective, take off the valve body and insert a blind gasket between it and the cylinder.

Carrying Water. Most of the later compound locomotives are equipped with piston valves, and it is very necessary that the cylinders should be kept free from water. Great care should be taken to open cylinder cocks when starting and before opening throttle after drifting down grade. Careful attention should also be given to avoid carrying water too high in boiler. Carrying water high in the boiler, and thus causing wet steam in cylinders, is injurious to compound locomotives, no matter whether slide valves or piston valves are used.

Oil Dash Pot. This should be kept full of oil, to prevent intercepting valve from slamming. Breakages of intercepting valves are nearly always due to neglect of this rule.

Dash pots should be filled with common car or engine oil, thinned with kerosene when necessary, in winter.

The dash pot stuffing boxes should be kept packed, to avoid leakage of oil.

Drifting. In drifting, the three-way cock should be in simple position whenever it can be done without too much loss of air by leakage of separate exhaust valve or piping. Most of the recent compound locomotives are provided with a small drifting valve, in main throttle

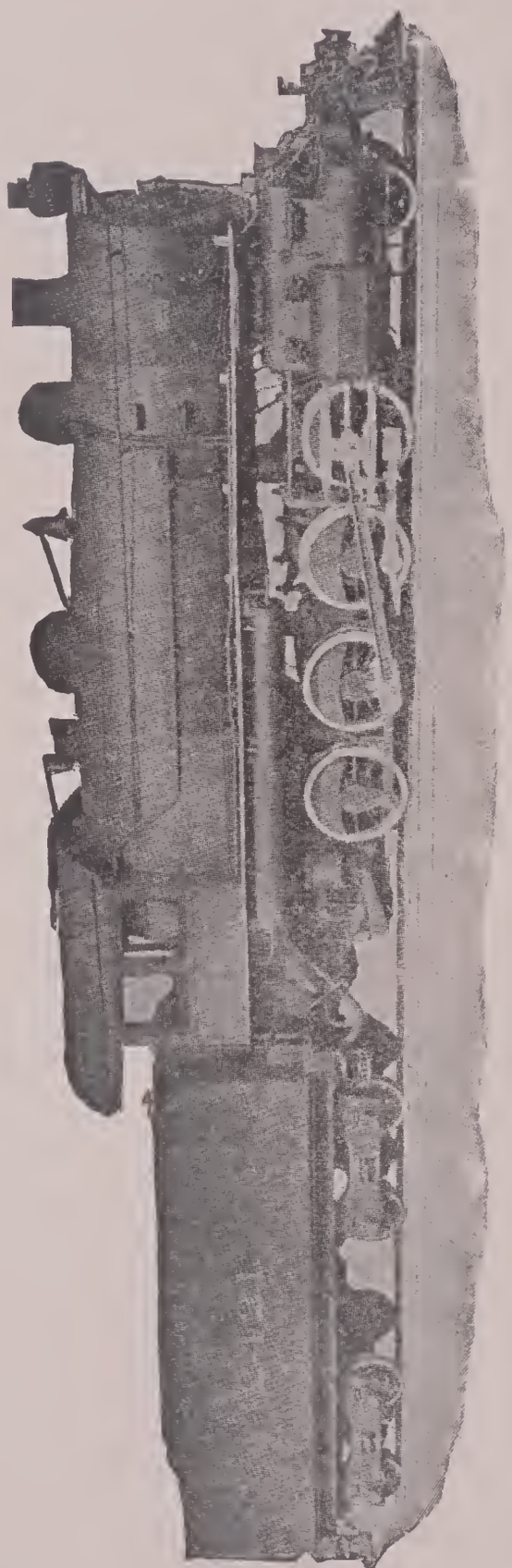


Fig. 177. Tandem Compound, Built for N. Y. C. and H. R. R. R.

Fig. 178. Four Cylinder Articulated Compound, Built for Baltimore and Ohio R. R. Heaviest and Most Powerful Locomotive Ever Built.

valve, so arranged that it can be opened with a slight movement of the throttle lever. It is considered good practice to admit a little steam to cylinders when drifting, through this valve, or, if not provided with a small drifting valve, by a slight opening of main throttle.

Examination. Enginemen should ascertain if separate exhaust valve is in good working condition before starting out with train, by trying the engine simple and compound before coupling to the train. The separate exhaust valve should be examined at intervals, so that the spring and other parts are kept in proper condition. Should the engine refuse to move after the throttle is opened, it will usually be found that it stands on center on high-pressure side (in position to take steam on low pressure side), and it will be due to either the intercepting or reducing valve sticking, which is always the result of lack of lubrication for intercepting valve, or carrying too much water in the boiler. Which of these valves are sticking can be ascertained from the position of the intercepting valve stem. In starting the engine, if the intercepting valve stem extends clear out about 7 in., it would be the intercepting valve, and unless some of the ports are broken a slight tap on the end of the stem, with throttle open, would send it ahead. If it was found that the stem had already moved ahead so that it extended out about 3 in., it would be the reducing valve. Usually one or two sharp blows on the intercepting valve back head, with throttle open, will loosen it. In either case live steam would then be admitted to low-pressure cylinder for starting.

Should the engine refuse to work compound after the three-way cock had been placed in compound position, and continue to work as a simple engine, it would indicate

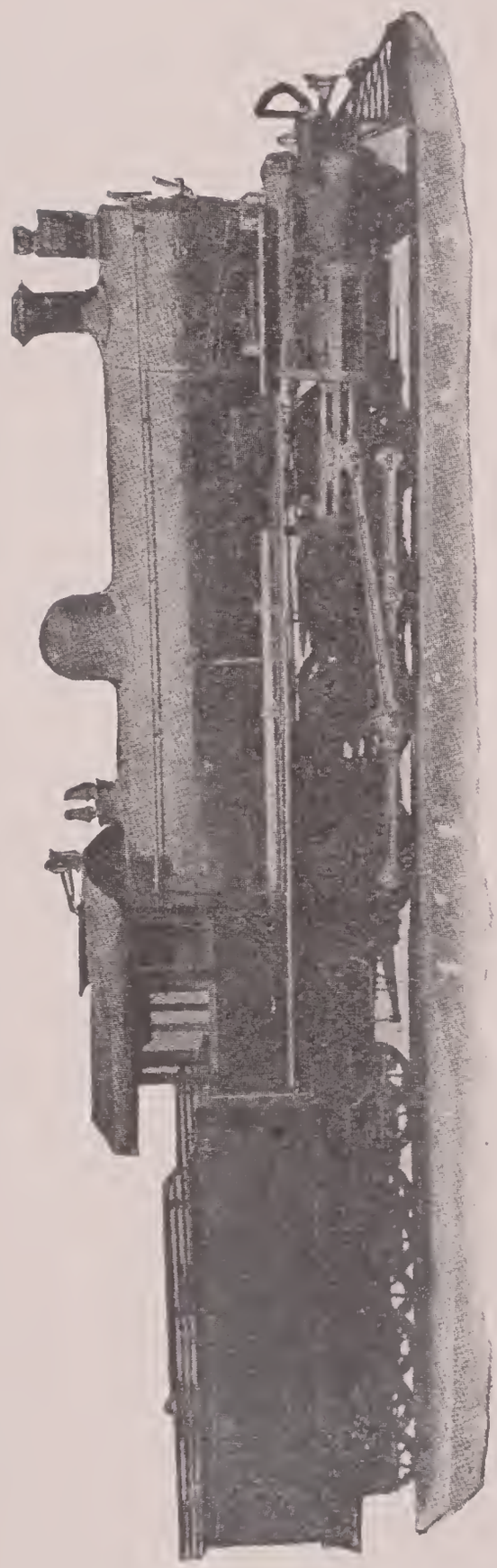


Fig. 179. Tandem Compound, Built for Cape Government Railways, South Africa.

that the separate exhaust had not closed. This trouble can usually be traced to enginemen using engine oil for lubricating separate exhaust valve chamber, and can sometimes be overcome by a dose of kerosene, which should in all cases be followed up with valve oil.

Relief Valves. Combined pressure and vacuum relief valves on low-pressure steam chest and single-pressure relief valves on low-pressure cylinder heads should be set at 45 per cent of the boiler pressure, and the high-pressure cylinder head relief valves set at 20 lbs. above boiler pressure.

Dampers. Dampers should be closed when drifting down long grades.

FOUR-CYLINDER BALANCED COMPOUND 2-6-2 AUSTRIAN LOCOMOTIVE.*

The latest compound locomotives designed by Herr Karl von Golsdorf for the Austrian State Railways are notable for the reason that they are the first engines for main railways either in Great Britain or the European continent to have the 2-6-2 or "Prairie" type wheel arrangement, and they are, besides, thoroughly up-to-date representatives of European compound locomotive design. According to the Golsdorf four-cylinder system there are two low-pressure cylinders outside the frames, and two high-pressure cylinders inside, all four cylinders driving the center pair of coupled wheels. As in other Golsdorf compound locomotives, no starting valve is employed, additional ports being fitted instead, in the working faces of the slide valves of the low-pressure cylinders, but it is arranged that, except in full forward or back-

*From the Railway and Engineering Review.

ward gear, these ports are never uncovered by the slide valves. In order to work non-compound, as at starting or to meet temporary requirements, it is only necessary

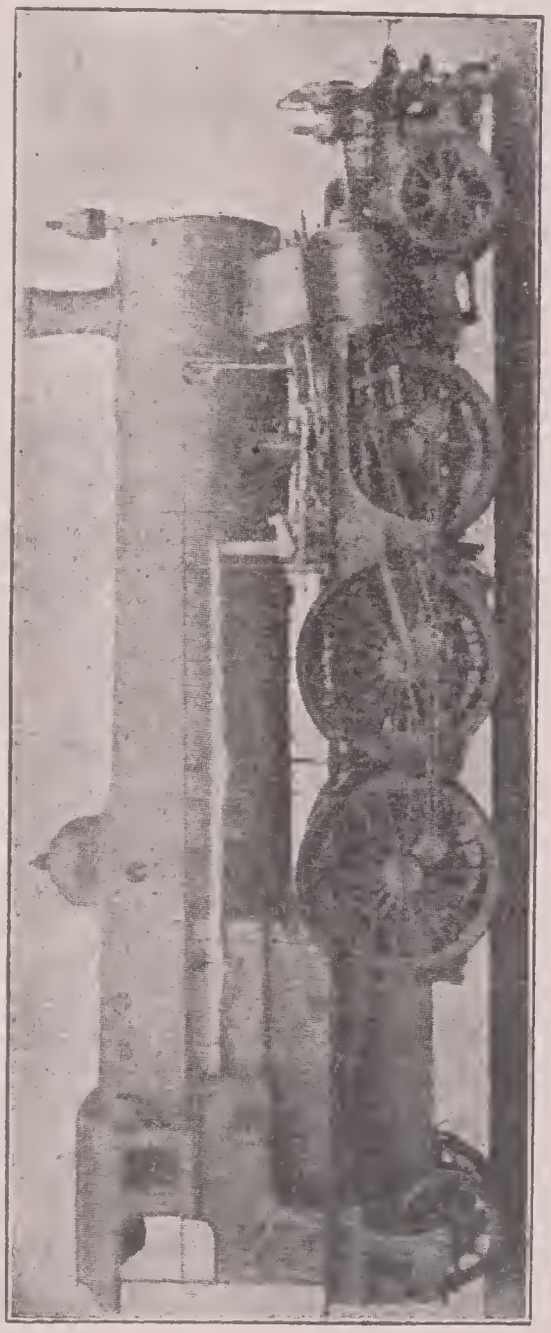


Fig. 180. Four-Cylinder Balanced Compound Locomotive for the Austrian State Railways.

to throw the reversing gear fully over, which is, of course, a natural operation with most drivers when great power is required, but as soon as the driver "hooks up" the engine works compound.

The boiler is a very large one, and is peculiar in that it is of the taper or wagon-top design, although very common in American practice, is not as yet very frequently employed in Europe. A wide firebox spanning and resting upon the frames is also included in the design. The valve gear is of the Walschaert type, two sets only being employed for the four cylinders, and the valves being placed above the cylinders, inclined in the case of those outside. Tail-rods are employed for all cylinders. Both the leading and trailing trucks are adapted for limited radial movement.

These engines are intended for working the very heavy express traffic over the steeply graded sections, ordinary 4-4-0 and 4-4-2 engines being used for the easier sections, and they have the following leading dimensions:

Diameter of H. P. cylinder, 14½ inches, diameter of L. H. cylinder, 24¾ inches; stroke, 28¼ inches; diameter of coupled wheels, 6 feet; diameter of leading and trailing wheels, 3 feet 5 inches; heating surface (total) 2,775.5 square feet; grate area, 43 square feet; boiler pressure, 213 pounds per square inch; weight on drivers, 42¾ tons; total weight, 69 tons.

FOUR-CYLINDER COMPOUND LOCOMOTIVE FOR THE HUNGARIAN STATE RAILWAYS.

The illustrations herewith are of a four-cylinder compound locomotive for the Hungarian State Railways, constructed at the Hungarian State Railways Locomotive Works, Budapest, and is one among many interesting exhibits at the Milan Exhibition. The following, together with the illustrations, is reproduced from Engineering, London, England:

. . . . The frame-plates are inside the wheels. The

two high-pressure cylinders are inside the frames, while the two low-pressure cylinders are outside. The cranks for the high, and low-pressure cylinders on one side of the engine are at an angle of 180 degrees, and the corresponding pair on the other side are also at 180 degrees, but the cranks of the right side lead by a right angle. All cylinders have piston-valve distribution. The low-pressure pistons are fitted with through piston rods. The boiler is provided with a wide firebox. The trailing axle is allowed radial play, and is brought back to its normal position on leaving a curve by spiral springs; laminated springs act in a similar way on the bogie. . . . .

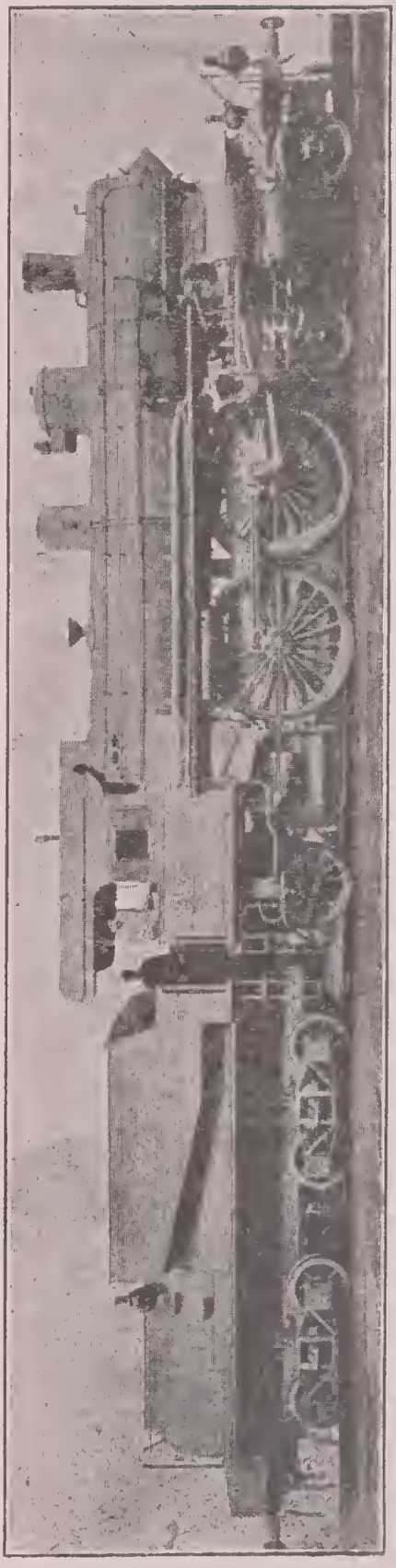


Fig. 181. Four-Cylinder Compound Locomotive for the Hungarian State Railways.

Weight of locomotive in working order, 74.3 tons; weight available for adhesion, 31.6 tons; weight of locomotive empty, 67.0 tons.

The firebox stays are of copper and phosphor-bronze. The boiler tubes are weldless; the stay tubes are 6 mil-

limetres (0.23 inches) in thickness, they are screwed and expanded in the fire-box tube-plate, and are held in the smoke-box tube-plate by nuts and washers on both sides. In the fire-box casing there are provided twenty mud-holes distributed over the sides and ends; there are besides in the fire-box foundation ring pipe cleaning-out holes, closed with screw plugs. The first and third rings forming the barrel are also each provided with a mud-hole, and two others, closed with screw plugs, are cut in the lower part of the smoke-box tube-plate. The man-hole is in the top of the third ring. The dome is mounted on the first ring of the boiler barrel, and contains a steam-separator and the regulator, the latter is lubricated by the oil-feeder in the steam-dome cover.

The casting forming the high-pressure cylinders is bolted to the boiler; provision is made for the free expansion of the latter at the back end. The longitudinal seams of the barrel are butt-jointed with double straps, and triple riveted. The boiler is fed by two Friedmann restarting injectors, of 11 millimetres (0.43 inches) bore. They are operated by steam led through a pipe from the steam-dome. A branch on the left-side feed-pipe is used for sprinkling the coal; the right-hand side feed-pipe is fitted with a cock for connection with the ash-box jet. The latter feed-pipe is also provided with the usual screw-joint for coupling with the fire-extinguisher hose. The fire-box has the usual fire-brick arch. In order to ascertain the extent of the air draught in the smoke-box when required, a copper pipe is fitted between the smoke-box and the cab, ending at the latter in a U-shaped glass indicator.

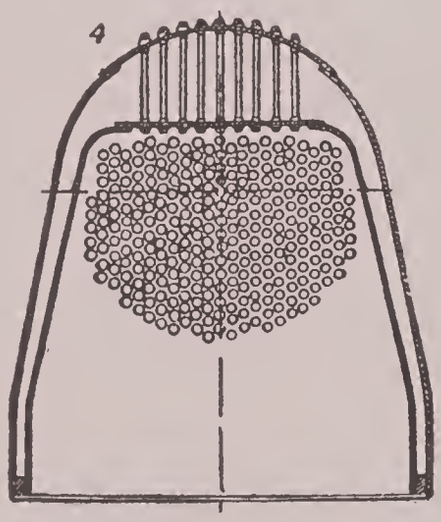
The steam is led from the regulator to the high-pressure cylinders, located, as above stated, inside the

main frame; from the high-pressure cylinders it passes to a two-way device connected with the low-pressure cylinders, thence either into the latter cylinders or to exhaust. On starting, and until the engine has reached full speed, live steam is supplied to all four cylinders; in this case the steam-valve of the two-way device is maintained open, and the steam from the high-pressure cylinders flows to exhaust, a branch on the high-pressure cylinder steam-pipe supplying live steam to the low-pressure cylinders direct. For compound running the two-way valve is closed, and communication is established between the high and low-pressure cylinders.

The valve-gear is on the Heusinger system. The link motion is outside the frame; the valves of the high-pressure cylinders being operated by the outside link motion by means of a double lever connected to the outside valve-rod. Lubrication of the piston-valves and cylinders is secured by two Friedmann lubricators. For running without load, air-valves are provided for the high and low-pressure cylinders, which communicate with the steam-chests. The air-valves of the low-pressure cylinders communicate by a copper tube with a pressure gauge in the driver's cabin. Both valve-chests of the low-pressure cylinders are provided with a relief valve, set for a 7-atmosphere (100 pounds per square inch) pressure.

All the axles, with the exception of the driving axle, are bored through. The crank-pin of the driving-axle is also bored. The sand-box is fitted with a compressed-air device. All the wheels are fitted with the Westinghouse brake; the brake-shoes of the rear trailing axle are carried by a universal joint suspension. The bogie-wheel sets are provided with a vertical brake cylinder 10 inches in diameter; for the other wheels there are two similar cylinders 13 inches in diameter.

The boiler barrel is of mild steel with a breaking stress of 36 to 42 kilogrammes per square millimetre (22.9 to 26.7 tons per square inch), and an elongation of 22 per cent.; the breaking strain and elongation figures together must give a minimum total of 61. The driving axle is of nickel steel, having a breaking stress of 55 to 65 kilogrammes (34.9 to 41.3 tons per square inch) and a minimum contraction of area of 40 per cent. The tires of the locomotive and tender wheels are of nickel-chrome steel; the axle-boxes of the trailing-axle and tender-axles are steel castings.



Section through Firebox

Fig. 182.

The tender-tank is cylindrical, with the exception of the front top part, where the water-tank is cut slanting to form the inside of the coal-bunker. The water-tank is fitted to two longitudinal beams which extend to the rear and front, and act as stays for the buffer beams. The tender is carried on two bogies. The bogie frame consists of a cast-steel cross-bearer of U section fixed at both ends to the axle-box framework; this is built up of flat steel bars, and contains the bearing springs as shown.

Both hand and Westinghouse brakes act on all the wheels. The brake cylinder is horizontal, 10 inches in diameter. In the rear of the coal-bunker the water-tank is provided with four holes for filling; which serve also as manholes. A gauge-glass is fitted to the front of the water-tank; connections are also provided at the rear end for the fitting of a second water-gauge glass should this be required. The water-tank is divided inside by cross partitions.

The cross-bearers of the tender-bogies are steel castings, the breaking stress of which is 40 to 50 kilogrammes (25.4 to 31.7 tons per square inch), with an elongation of 15 per cent.

During tests made with this locomotive on the line from Budapest to Vienna a load of 75 tons was hauled on a level at a speed of 142 kilometres (88.5 miles) per hour. With a load of 357 tons the speed was 110 kilometres (68.5 miles) per hour.

FRENCH 6-2-2-6 COMPOUND FREIGHT LOCOMOTIVE.

The Nord Railway of France has a very large passenger traffic out of Paris, bound to and from London, as well as to the Belgian frontier, and in addition is regularly called upon to handle a very extensive coal traffic from the Lens district and the northeast of France generally. For this coal traffic, a number of powerful six-wheeled coupled compound locomotives have, within the last few years, been introduced on the Nord Railway, but although these machines have been capable of hauling about 1,060 tons, on the section of railway between Lens and Paris-La Chapelle, at a speed of over 20 miles per hour, they have not been found altogether suitable

for taking the heaviest loaded freight trains over the stretch of track from Hirson to Valenciennes and Busigny, where the gradients encountered are somewhat unfavorable. As a result the heaviest of the coal trains from the Lens basin have, upon arrival at Valenciennes and Busigny, had to be split up and additional locomotive power requisitioned from these points. With a view to reducing the cost of this locomotive power and to operate the coal traffic with increasing economy, Mons. du Bousquet, the chief mechanical engineer of the Nord of France Railway, has quite recently designed an entirely new type of locomotive, one of which was on exhibition at the Liege Exposition, whilst the other engine has been put to actual work in hauling heavy coal trains, and is, by the courtesy of Mons. du Bousquet, illustrated in the accompanying illustrations. As will be noticed the two sets of wheels of the locomotive are independent of each other, and, in addition to six coupled driving wheels, each has a pair of carrying wheels in front of the cylinders. The driving wheels are each 57¼ inches in diameter and the carrying wheels 33½ inches in diameter. The locomotive is of the four-cylinder compound type—the high pressure cylinders, being 15¾ in. in diameter with a 26¾-in. stroke, drive the rear set of wheels, whilst the low-pressure cylinders, which are 24¾ inches diameter, with a piston stroke of 26¾ inches, actuate the six coupled wheels of the front set. Both of the running gear pivots are joined by a central built-up girder, or main underframe, which also carries the boiler, side tanks and other appendages. Where a large tractive force is desired, each pair of cylinders can receive live steam, thereby increasing the theoretical force from 18½ tons to 24 tons.

The steam passes from the boiler to a rotary joint

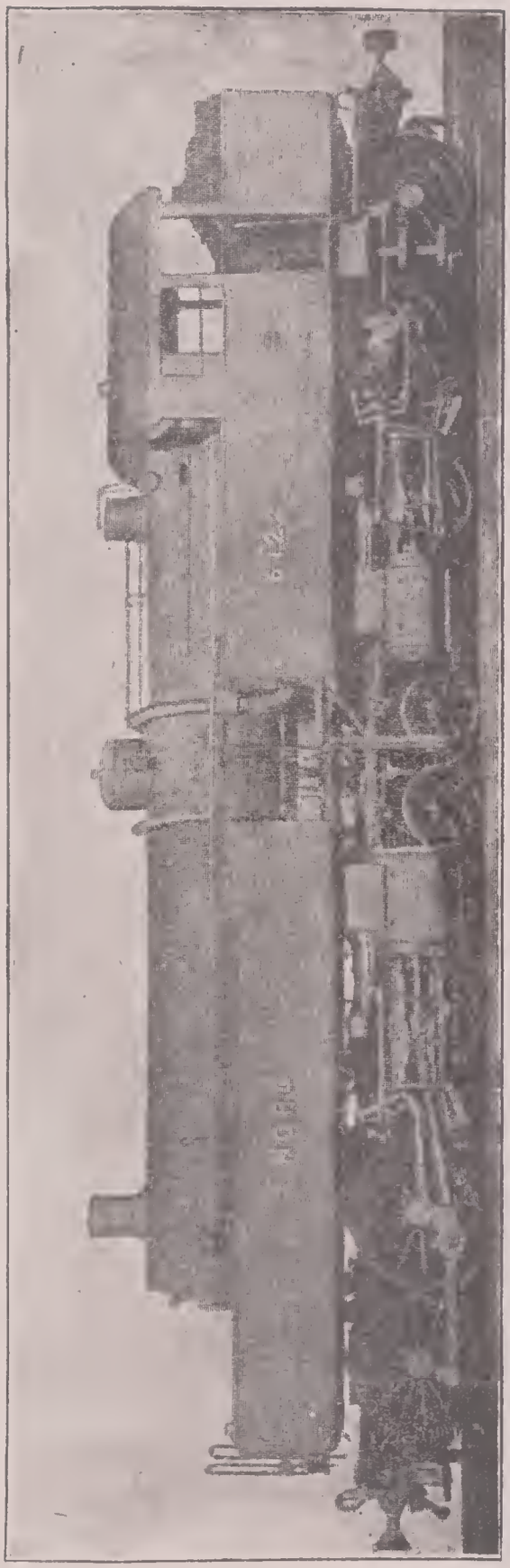
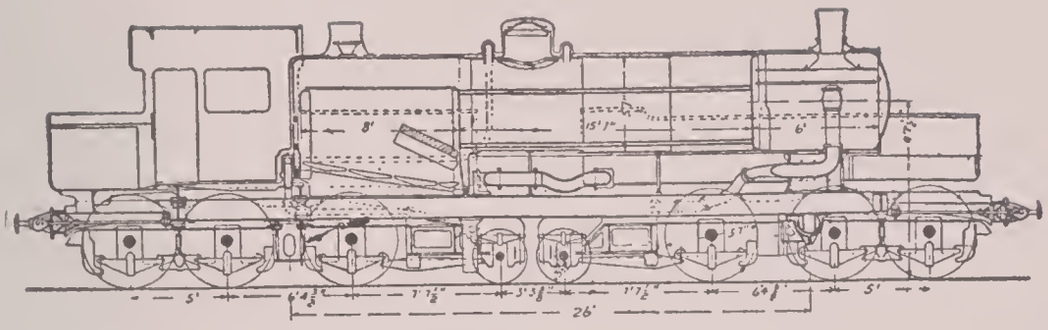


Fig. 183. French Compound Duplex Locomotive, Nord Railway.

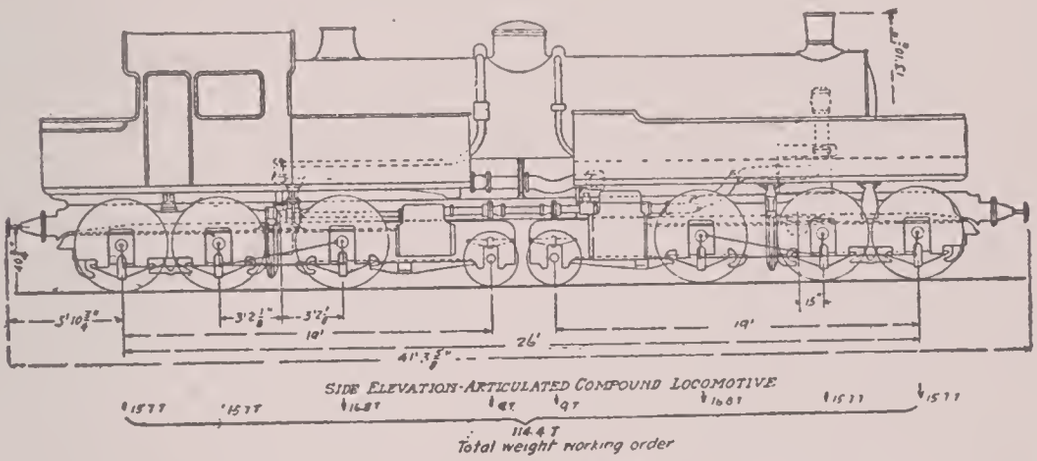
at the rear of the firebox—below the fire door and above the pivot of the rear set of wheels. From this rotary joint, a separate steam pipe supplies steam to each of the high-pressure cylinders, a special system of knee-jointed pipes connecting the high-pressure cylinders with the low-pressure cylinders, whilst the utilization of the new type of intercepting, and starting valve recently brought out on the Nord of France Railway permits of the starting of the two groups of engines as simple expansion engines. The ends of the two cross beams are provided with slides, bearing on castings attached to the rear running gear frames, and by this means the rear set of wheels can turn in a horizontal plane around its pivot, but can not move in any other direction, as can the front set, which have a spherical pivot. Buffer springs are provided, with a view to minimizing the transverse oscillations, and the front set can thus adapt itself to the inequalities of the road, without in any degree affecting the suspended mass resting on the central girder. In order that the front wheels may be sufficiently loaded, a portion of the water tanks is arranged on this end, this tank having a carrying capacity of 1,980 gallons, sufficient space having been allowed between the boiler and tanks to permit of the angular movement of the front wheels and frame. The remainder of the water tanks are located on either side of the firebox, and there is accommodation here for 836 gallons.

The boiler is of somewhat unusual dimensions, its barrel having a length of 15 ft. 10¼ in. and a diameter of 4 ft. 9¼ in., while its center line is 9 ft. 2¼ in. above rail level. The high pressure, and low pressure machinery is similar.

The slide valves are of the usual D type, and balanced,

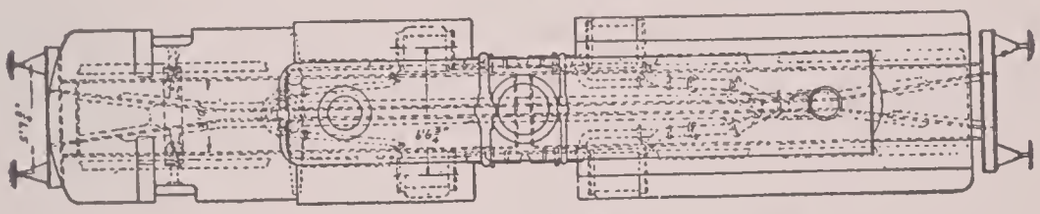


LONGITUDINAL SECTION ARTICULATED COMPOUND LOCOMOTIVE NORD RY FRANCE

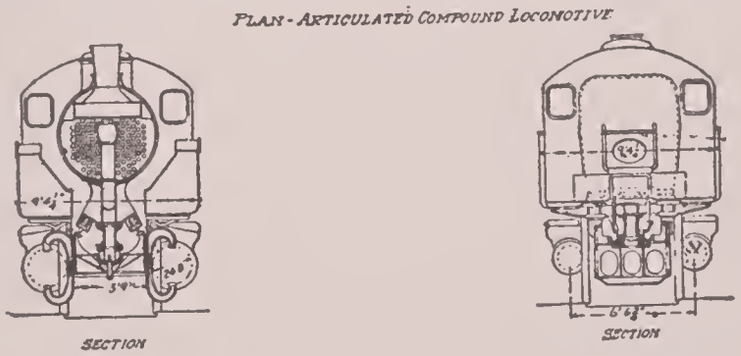


SIDE ELEVATION-ARTICULATED COMPOUND LOCOMOTIVE

114.4 T
Total weight working order



PLAN - ARTICULATED COMPOUND LOCOMOTIVE



SECTION SECTION

Plan, Elevations and Sections, Compound Duplex Freight Locomotive
Fig. 184.

and the axles and crank pins are of nickel and chromium steel.

The centers of the pivots between the main frame, and the independent frames for each set of wheels are 26 ft. 10 in. apart.

The total heating surface is 2632.36 sq. ft. to which the firebox contributes 129.06 sq. ft. and the tubes which are of the Serve pattern and are 130 in number, contribute 2503.3 sq. ft., the grate area being 32.29 sq. ft.

There is space at the rear for five tons of coal. In running order these four cylinder compound duplex freight locomotives have an aggregate weight of no less than 114.4 tons, although, it is interesting to note that the maximum weight on any pair of wheels is only 16.8 tons. In the course of some recent official trials, the engine illustrated herewith proved its ability to haul a load of nearly 900 tons, over a grade rising 1 in 80, at a speed of 12 miles per hour, the engine passing with ease over curves of 231 feet radius, whilst when it was traveling at a speed exceeding 43 miles per hour, the engines were perfectly steady. Considerable interest is being taken in Great Britain, and also in Continental circles, in the working of these two locomotives, as should the expectations of Mons. du Bousquet be realized, there is liable to be something akin to a mild revolution in steam locomotive design for heavy work.

Questions.

428. What is the principal object in the compounding of locomotives?

429. Name two sources of economy in compound engines.

430. Why is there a constant loss of heat in the single cylinder engine?

431. How is the expansion of the steam divided in the compound locomotive?

432. How should the cylinders of a compound engine be proportioned regarding size?

433. What other problems are before the designers of compound locomotives?

434. How many types of compound locomotives are in use in this country?

435. Describe briefly the Vauclain compound.

436. What kind of an engine is the balanced compound?

437. How are the cylinders of the tandem compound located?

438. How many cylinders has the cross compound?

439. What kind of valve gear is used on compound locomotives?

440. What were some of the objects aimed at in designing the Vauclain compound?

441. How many and what type of valves are used on the Vauclain compound?

442. What kind of packing rings are used on this valve?

443. When is the Vauclain valve motion direct acting?

444. When is it indirect?

445. In setting these valves, what ports are to be considered?

446. Of what material are the pistons made?

447. In starting these engines with full trains, what is necessary?

448. How is this accomplished?

449. What is the starting valve, and what is its function?

450. How is it operated?

451. What rule should be observed regarding this valve?

452. What provision is made for taking care of water that finds its way into the cylinders?

453. What is the first thing an engineer should learn, in the operation of a compound locomotive?

454. How is the quadrant of the Vauclain compound made, with references to point of cut-off?

455. What rules should be observed when starting the Vauclain compound?

456 When should the reverse lever not be hooked up?

457. Should the starting device be used when the train is in motion?

458. When is it allowable to use the starting device while the train is in motion?

459. How should the fire be carried in the Vauclain compound?

460. Where is the most economical point of cut-off for a single expansion engine?

461. Where is the most economical point of cut-off for a compound locomotive?

462. What should be done when starting the Vauclain compound?

463. What should be done with the reverse lever as the speed of the engine increases?

464. What should be done on a slightly descending grade?

465. What should be the position of the starting valve lever, when throttle is closed?

466. If there is danger of stalling on a heavy upgrade what should be done?

467. What is one of the legitimate advantages of the compound locomotive?

468. What advantage has the boiler of a compound locomotive over the boiler of a simple engine?

469. What is the ideal type of engine, whether stationary or locomotive?

470. How may this ideal be reached?

471. What has always been a serious problem for locomotive builders?

472. How are the cylinders of the Baldwin balanced compound located?

473. What type of valve is used on these engines?

474. Where are the valves located?

475. Where are the high-pressure cylinders located?

476. At what angle are the cranks set?

477. Describe briefly the action of the steam in this engine.

478. How are the cylinders of the American Locomotive Company's balanced compound located?

479. How is a uniform turning moment attained in this engine?

480. Mention the advantages that the balanced compound possesses over other types of compound locomotives.

481. Why does the tandem compound appear to be the ideal locomotive?

482. What is one of the main objections to this type of compound locomotive?

483. What kind of handling does a compound engine require?

484. What knowledge is necessary for the engineer in order that he may successfully operate a compound engine?

485. What can be said regarding the tandem compound built by the American Locomotive Co.?

486. How are the valves arranged on this engine?

487. What is the function of the starting valve?

488. How should a compound locomotive be lubricated?

489. How are the cylinders placed in the Baldwin tandem compound?

490. What about the cylinders and valve chests of this engine?

491. What kind of a valve has this engine?

492. How many cylinders has a cross compound, and how are they located?

493. What is the purpose of the intercepting valve?

494. What is the function of the automatic reducing valve?

495. How is the steam for operating these valves supplied?

496. How is the receiver pressure governed?

497. What are the by-pass valves for?

498. What precautions should be observed regarding water on these engines?

499. What about the oil dash pot?

500. What should be done when drifting?

501. What should be done with the separate exhaust valve?

502. Should the engine refuse to move when the throttle is opened, what would be the probable cause?

503. How may it be ascertained which one of these valves is stuck?

504. How may the stuck valve be loosened?

505. In what position should the dampers be when drifting?

THE ELECTRIC HEADLIGHT.

A good headlight is a very necessary part of the equipment of a locomotive. It matters not how well the engine and train may be equipped with the air brake, and other safety devices, if the man at the throttle does not or cannot see the danger ahead in time to apply the "air" and make the stop, the result is going to be a disaster of a more or less serious nature. Of course, by daylight there is practically no excuse for the engineer not seeing what is in front of him, but in hours of darkness the situation is reversed, and unless the engine is equipped with a headlight that will project the light a far enough distance ahead to enable the engineer to see the impending danger in time to stop, or at least to considerably slacken the speed of the train, the responsibility for a smashup should not rest wholly upon the shoulders of the engineer. He may know every foot of track from one end of the division to the other end, but he cannot know intuitively whether or not the track ahead of him is clear and safe, except he can see it.

The ordinary oil lamp falls far short of the requirements of a good headlight, and, until inventors shall succeed in perfecting a headlight that will illuminate the track ahead of the engine for a sufficient distance to cover the number of yards distance required in which to make a stop, the problem of the safe running of trains at night will remain in an unsolved state. There is but one device that appears to offer a solution of this most important question, and that is the electric head-

light. Inventors have been striving for quite a number of years to bring out a reliable and efficient light of this kind, and although the process of development has been slow, still progress has been made, especially since the steam turbine has become available for operating the small dynamo, or generator required to furnish the electric current needed. There is no doubt that the electric headlight will in time take its place as an element of safety, second only to the air brake in the running of railway trains. In order that the reader may get at least an idea of what is being accomplished along this line, the author presents the following information, descriptive and otherwise, together with illustrations.

PYLE ELECTRIC HEADLIGHT.

The Pyle National Electric Head Light Company of Chicago are the makers of a light that appears to be deservedly working its way to the front, there being now about five thousand of their headlights in use, with a growing demand for more.

The device may be located either on the front end of the engine forward of the stack, or the generator may be located on the left side just in front of the cab, while the lamp will occupy the usual position in front of the stack. Figure 185 shows the complete outfit on the front end ahead of the stack, and figure 186 shows the generator located on the boiler just in front of the cab.

Instructions for Applying the Equipment. If it is to be placed on the front end as shown in Fig. 185, measure the depth of the headlight case to be used and add 18 inches to this measurement. Then have a base-board made, using these measurements for the length of board,

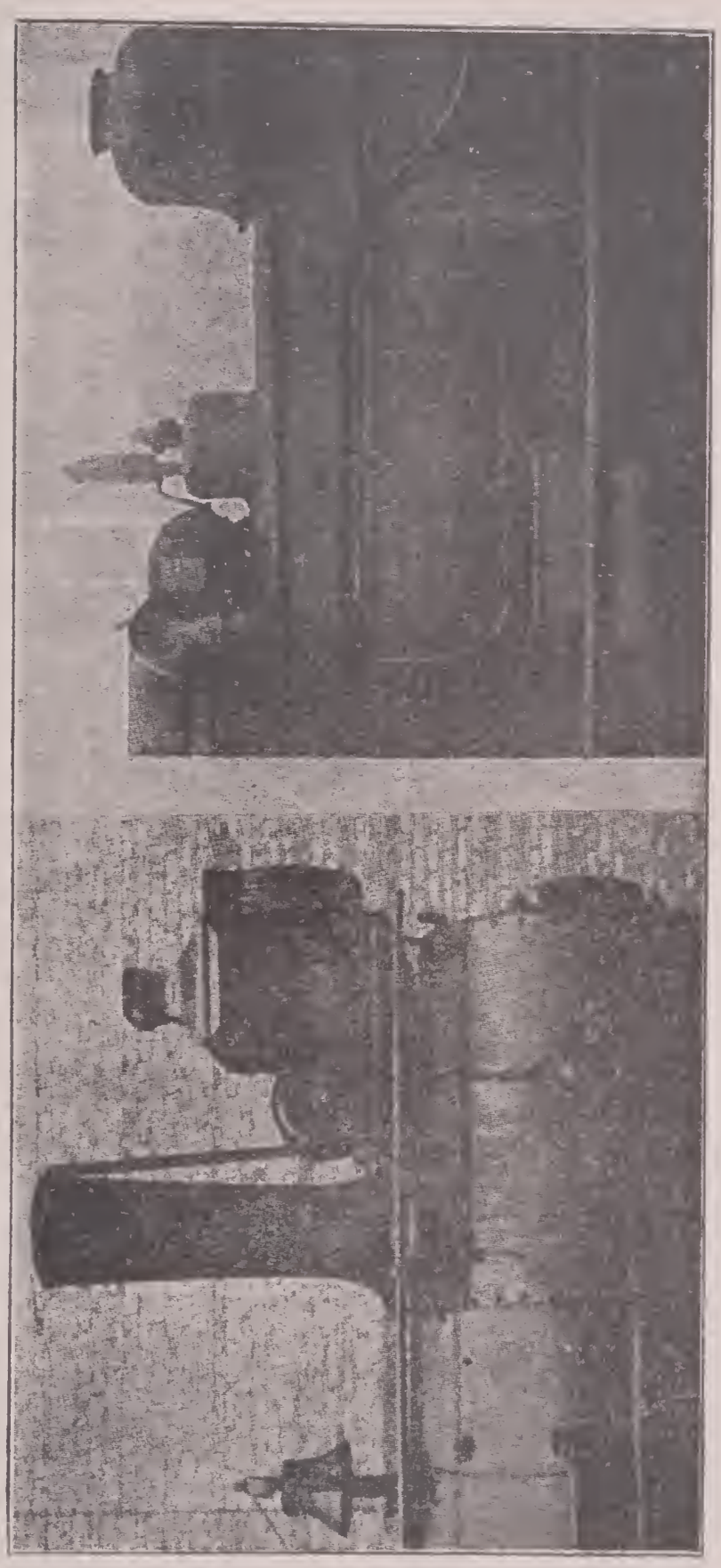


Fig. 186.

Fig. 185.

and make the breadth the standard width for oil headlights. Bend a piece of ¾ by 3 in. iron flat ways "U" shaped, making center of iron come over the center of brackets on engine and make the length about two inches shorter than the base-board. Place this iron "U" on the brackets within about two inches from the stack and mark for bolts through brackets.

Put base-board on the iron "U" so the back edge will just clear the stack, and bolt to brace. Place the equipment on the board, with dynamo on the left side of locomotive and as near the back edge of board as possible, and bolt to board. Then bolt the headlight case to board in front of equipment. Put a ¾-in. angle valve in the highest part of the boiler in the cab with dry pipe, and run a ¾-in. pipe from this valve under the jacket on right side of the boiler to the equipment. Use 1½-in. pipe for the exhaust, running it through the arch, and have end of pipe about flush with the top of the nozzle tips of the locomotive. Use 2-in. pipe inside the arch, as it reduces noise and does not act on the fire.

If the equipment is to be located on the boiler just in front of the cab, take up back sheet of jacket and fasten two brackets to boiler suitable for holding the equipment crosswise of boiler. Have a base-board (iron or wood) made and bolt to braces, then place equipment with the dynamo on left side of locomotive and bolt to base.

If steel or iron base-board is used, the equipment should be insulated from base by wood or asbestos sheet. Place a ¾-in. angle valve in the highest part of boiler in the cab (with dry pipe) and run a ¾-in. iron pipe to the equipment. Run the 1½-in. exhaust pipe up, above cab or down, to running board and into arch. Run the lead

wires through an independent pipe if possible, or through the hand railing, being sure to insulate them carefully to prevent chafing, etc.

To apply the lamp to the reflector and case, remove the oil tank, and all supports and guides from the reflector. Cover the board holding the reflector with tin, having the edges turned up about one-half inch.

This prevents sparks from the lamp from setting fire to the oil-saturated board. Secure the support for the back of the reflector to the lamp board on righthand side, and adjust the screw until the reflector stands level. Remove both carbon holders from the lamp and set the lamp on the board, on the side nearest door of case, then put bottom holder in lamp and place the lamp on the board with copper electrode in center of hole in reflector, being sure to have base of lamp square with reflector slide. Mark for holes in center of large square holes in base of lamp and bore for ⅜-in. bolts.

Secure the small wires for the incandescent lamps by the small screws at the right of brush holders. Run the incandescent wires through the hand railing to the cab, in wood strips. Run wires for cab in wood strips. Wind all the joints with tape.

The electric headlight equipment complete consists of engine, dynamo and lamp, and material for three incandescent lights in cab.

The Engine. The engine is known as the Pyle compound steam turbine. There are no wearing surfaces inside the engine requiring lubrication, hence they do not use any sight feed lubricator in the cab. Before starting the engine be sure the casing is thoroughly drained, and do not turn on steam too suddenly in starting the light, thus allowing time for the condensation to get out of the engine. It must have dry steam.

Remove plug in top of engine once each week and pour in a little black oil. This will prevent corrosion of parts. The inside bearing only needs enough oil in the well for the loose ring to touch the oil and carry up on top of the shaft. If there is too much oil, it will be thrown out of the ends of the cellar by the motion of the locomotive, which may ruin the armature. The oil well for the outside bearing should be filled each trip. Use valve or cylinder oil in these bearings.

The Dynamo. The dynamo is constructed on the latest scientific principles, and the electrical balance is so perfect that no sparks should be seen at the brushes. The armature is held in place on the engine shaft by one screw which can be easily taken out if occasion demands. The brush holders are fixed, and the brushes can be taken out and replaced without changing the tension of the springs. A graphite brush is used for the top and a carbon brush for the bottom, and a few moments' care each trip is all that is required.

The mica between the copper strips of the commutator should always be a trifle below the surface. If it gets too high, file it down with a small file. Do not get it too low, as it will collect dirt, etc., and cause a short circuit. The commutator should be cleaned each trip with a damp piece of waste (not wet), rubbing endwise so as to keep the creases clean where mica is filed out.

Be sure and have the brushes fit perfectly on the commutator. If there is poor contact brushes will spark. If commutator is running out or has the appearance of getting rough, clean it up. To do this nicely, remove brushes and hold a strip of No. 0 sandpaper (not emery paper) on the commutator while run-

ning (see Fig. 187). Don't press the sandpaper on, for if there are any low spots they will increase in size.

If the brush tension spring is too tight, it creates friction, heat and unnecessary wear, both on the commutator and the brushes. If too loose it will spark and commutator will not run clean. Have it just tight enough to prevent sparking. In this case a little judgment must be used, for if the brushes are not in the proper condition, or commutator smooth and true, there will be sparking at the brushes, no matter how much

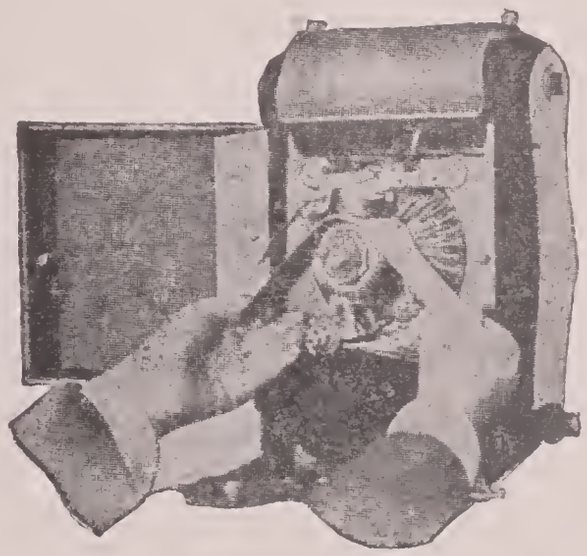


Fig. 187.

pressure is used. Do not forget that the commutator is the vital part of all dynamos, and none will run successfully without regular care and attention. The voltage of the dynamo is entirely too low to force a current through any portion of the human body, so it may be handled freely and without any possible fear of being injured by it. It only requires a few moments' attention each day to keep the plant in perfect condition.

If the commutator becomes rough or out of round, it should be trued up in a lathe. The tool used must

be very sharp, and light cuts must be taken, then polish it with fine sandpaper. It must be carefully examined to see that no two sections touch, as the copper is liable to lag or burr from one section to the other, and before putting it back, it would be better to cut or file the mica (between each section) a little below the surface, for it does not wear away as fast as the copper, and if the mica is not cut away it may lead to sparking. After doing this, be sure no ragged edges of copper stick up, for this will cut away the brushes rapidly. The speed

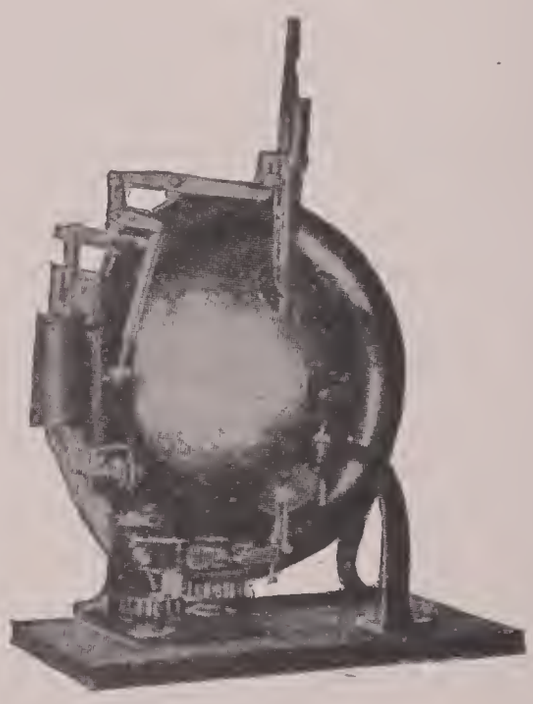


Fig. 188.

of the armature should be as near 1,800 revolutions per minute as possible, unless the copper electrode burns off, when it should be reduced until the copper electrode does not burn.

The Lamp. (Fig. 188)—is simple, durable and reliable, and after a few trials it becomes an easy matter to trim the lamp in the dark, should occasion demand.

In putting in the top carbon, it is much better to remove the carbon holder from the slide. After securing the carbon in the holder, take it between the thumb and forefinger and with the remaining fingers resting on the guide put it in place. If desired to clean the reflector remove only the top guide by loosening the thumb nut at the end of the upper arm, then remove the guide carbon and carbon holder.

The tension spring in the lamp is for two purposes. It brings together the points of the carbons, so as to establish the arc when the dynamo is set in motion, for there must be a complete circuit before there is any current. If the carbons are separated only a small fraction of an inch, the lamp will refuse to work, because the current will not jump across the separation. Sometimes there will be a deposit of scale on the point of the lower copper electrode which prevents the top carbon touching the copper, and as the current will not go through this scale, there will be no light until this is removed. See that the point of copper is clean before each trip. Suppose all wires are connected and the lamp properly trimmed, turn on steam and set the armature in motion. The current enters the lamp, and passing through or around the solenoid magnet, draws down the iron armature. This in turn separates the carbons, thus forming the arc or light. It will be noticed that the spring is secured to the end of the lever toward the carbons, or on the opposite end from the magnet and pulls against it. This prevents the solenoid from pulling the carbon too far apart. The volume of light will depend largely on the way this tension spring is regulated. It may be so tight that the magnet will be unable to separate the carbons, consequently there will be no light.

If the dynamo is run too long with the lamp in this condition it will burn out the armature or the fields, for the current becomes very heavy.

If the tension spring is very loose, the lamp will flash and go out, for the magnet will be drawn down too far. When the light goes out the current is broken, and there being no strength in the magnet, the spring will again bring the carbons together, then the current is instantly re-established. Adjust the spring so the lamp will flicker just a little, when the locomotive is at rest.

The wires leading back to the incandescent lamps may come together, causing a short circuit. This will put the light out. When this occurs the dynamo will be generating a heavy current, the speed will be quite low, and there will be a small light in the lamp. In this case just disconnect one of the small wires from connecting screw, then look for the cause of the trouble.

Most of the troubles are traceable to the adjustment of the lamp.

The magnet yoke may travel too far sometimes, and strike the small lug on frame of lamp before carbons are separated sufficiently to make a proper arc. In this case the wire should be shortened so that the magnet yoke is about half way down before the clutch grips carbon.

If the wire is too short the lamp will jump, or the carbon will stick in the clutch.

If the carbon feeds too fast the top clutch spring is too weak and should be given more tension. To do this, remove cotter pin and get top-clutch spring out of casing. Then pull it out a little, thereby giving it more "set."

If the light burns green, the dynamo is running too fast, and the speed should be reduced.

This can be stopped on the road at once by throttling the steam in the cab. There is another reason for light burning green. The main wires from the dynamo to the lamp may be connected wrong, therefore one wire should have a sleeve on each end large enough to prevent its going into the binding post with the small hole. The other wire should have plain ends.

The lamp can be moved in all directions for focusing. To get the proper vertical focus on the track, either to have the light close to you or to strike the track far ahead, loosen the set screw on the side, and by turning the adjusting screw the lamp can be raised or lowered as desired. To move it sideways, backward or forward, loosen the hand nuts and the lamp is free to move.

When once in focus there is no need of changing it again. Tighten all screws.

The back of the reflector is supported by an adjustable step, with a screw to raise or lower it, so the volume of the light will come out in parallel lines.

Focus Lamp. 1. Adjust back of reflector so front edge will be parallel with front edge of case.

2. Adjust lamp to have point of copper as near center of reflector as possible.

3. Have carbon as near center of chimney hole in reflector as possible.

4. Have locomotive on straight track and move lamp until best results are obtained on track. The light should be reflected in parallel rays, and in as small a space as possible.

To lower light on track, raise lamp. To raise light on track, lower lamp.

If the light throws any shadows it is not focused properly.

If light is focused properly and does not then strike center of track, do not change focus, but shift entire case on base board.

Point of copper should be about one inch above top of holder. If it is higher than this there will be too much heat on clutch.

To Roundhouse Men. A centrifugal brake is placed on back side of spoke of wheel and should be set so as to act at about 100 revolutions per minute more than where the governor acts, so that, if for any reason, the governor fails to act, this brake will check the speed and hold it at any speed at which the brake is set. The application of this brake commences with equipment No. 2,600, but cannot be applied to equipments with serial number lower than that.

To adjust centrifugal brake, remove the armature and cap to the engine, pull out wheel and shaft when there will be free access to brake. If it is desired to adjust brake so that it will act at a higher speed, turn nuts to right, being sure to adjust both brakes the same, then tighten up jam nuts. One-half turn of the nut will change the speed at which the brake will act about 150 revolutions.

The governor should be examined once each month, and if the plungers are found cut they should be ground in or faced off as the case requires. If plungers are cut, the engine may run away and be broken by centrifugal force. If plungers are faced off, the ends of governor yoke should be bent farther out from face of wheel, thereby allowing plungers to again seat firmly before governor weights are thrown out farther than at right

angles to face of wheel. If the speed is too high, adjusting screws should be turned back half a turn each, being careful to adjust all the screws the same. Half a turn of these screws should change speed about 100 revolutions per minute. If by turning back governor spring adjusting screw the speed is not reduced, the plungers do not seat, and should be faced off.

Suggestions. Have a few strips of No. 0 sandpaper about 1½ in. wide to clean up the commutator.

If the light fails to burn when turning on steam, see that all screws are tight, and that point of copper electrode is clean. Push down on lever and see if carbon lifts up and falls freely. Put a carbon across both binding posts, and if there is a flash when it is removed, dynamo is all right and the trouble is in lamp. If there is no flash when carbon is removed take out brushes and clean commutator with sandpaper (not emery paper), put the brushes back and try the carbon again.

Keep all screws tight.

After putting in a new carbon, always push down on lever, and notice if carbon lifts and falls freely. If it does not lift, it is not in the clutch. If it does not fall down freely, turn it partly around and find the freest place.

The carbon should burn from eight to nine hours.

Engineers should be held responsible for the proper care of the equipment unless some one is appointed to examine and care for them at roundhouses.

Before leaving on a trip the equipment should be started and brushes examined, as to tension of springs, and adjusted if necessary before getting out on the road.

These equipments are not automatic, and as there are quite a number of enemies to electricity on the locomo-

tive, such as grease, dirt, jar, heat, etc., it is necessary to give it a few minutes' attention each day.

Don't attempt to remove reflector from the case until the top carbon holder is removed by loosening thumb nut.

If the copper electrode burns off, equipment is running too fast, and the speed should be reduced by turning governor spring screws to the left until the trouble is stopped. Be careful and adjust all screws the same, as nearly as possible. One-half turn of screws will change speed about 100 revolutions per minute.

Be sure and adjust tension spring as loose as possible and not have the light go out while locomotive is standing still.

If the light dies down when locomotive is running fast, the tension spring may be too tight, which prevents solenoid from separating carbons sufficiently to form proper arc, or top-clutch spring may be too loose, allowing back edge of clutch to be jarred up and release carbon.

If the light goes out when the locomotive is standing still, the tension spring may be too loose or carbon may not feed freely.

If light burns green on the road, throttle steam at once.

If electrode does not come in line with the carbon, the holder should be bent until electrode comes in line with top carbon.

Both ends of one lead wire should be doubled about one inch, so it cannot go into binding post with the small hole, and thereby prevent crossing of wires.

Special motor brushes should be used with the improved brush bolders, top and bottom brush being of

the same quality. The graphite and carbon brushes used with the old style brush holders should not be used with the improved brush holders, nor special motor brushes in the old style holder.

EDWARDS ELECTRIC HEADLIGHT.

The Edwards Electric Headlight Co., of Chicago, whose factory is at La Porte, Ind., are manufacturing an electric headlight which, in addition to sending a powerful ray along the tracks in front of the engine, also projects a powerful vertical beam. The vertical beam makes a very decided illumination in the heavens, so much so that it is possible not only to detect the presence of an engine, but also in many cases to follow its path and determine in which way it is heading. An engineer is by this means placed in touch with the movements of other trains in his vicinity and is enabled to detect their presence where, if they carried ordinary horizontal beam headlights, he would be unaware of their location. The horizontal ray of light projected in front of the engine, highly illuminates the track and right of way for a distance of half a mile to a mile ahead of the train. Among the objections that have been raised to the use of electric headlights on locomotives is this one, that owing to the very brilliant light the engineer would not be able to distinguish the different colored signals, but practical experience seems to refute this idea, and the claim made for the Edwards light is that it does not in any wise decrease the efficiency of the signal lights, but on the contrary that they show up in their true colors.

Another valid objection is, that on a double track road there is danger of blinding an approaching engineer.

There is good ground for this latter objection and to guard against this contingency the apparatus is provided with a translucent shade, within the goggle, which may be drawn at will by the engineer when he is at the proper distance from an approaching engine. This shade destroys the strong glare of the light, giving the effect of frosted glass. As soon as the approaching train is passed the engineer releases the shade and again gets the full value of the light.

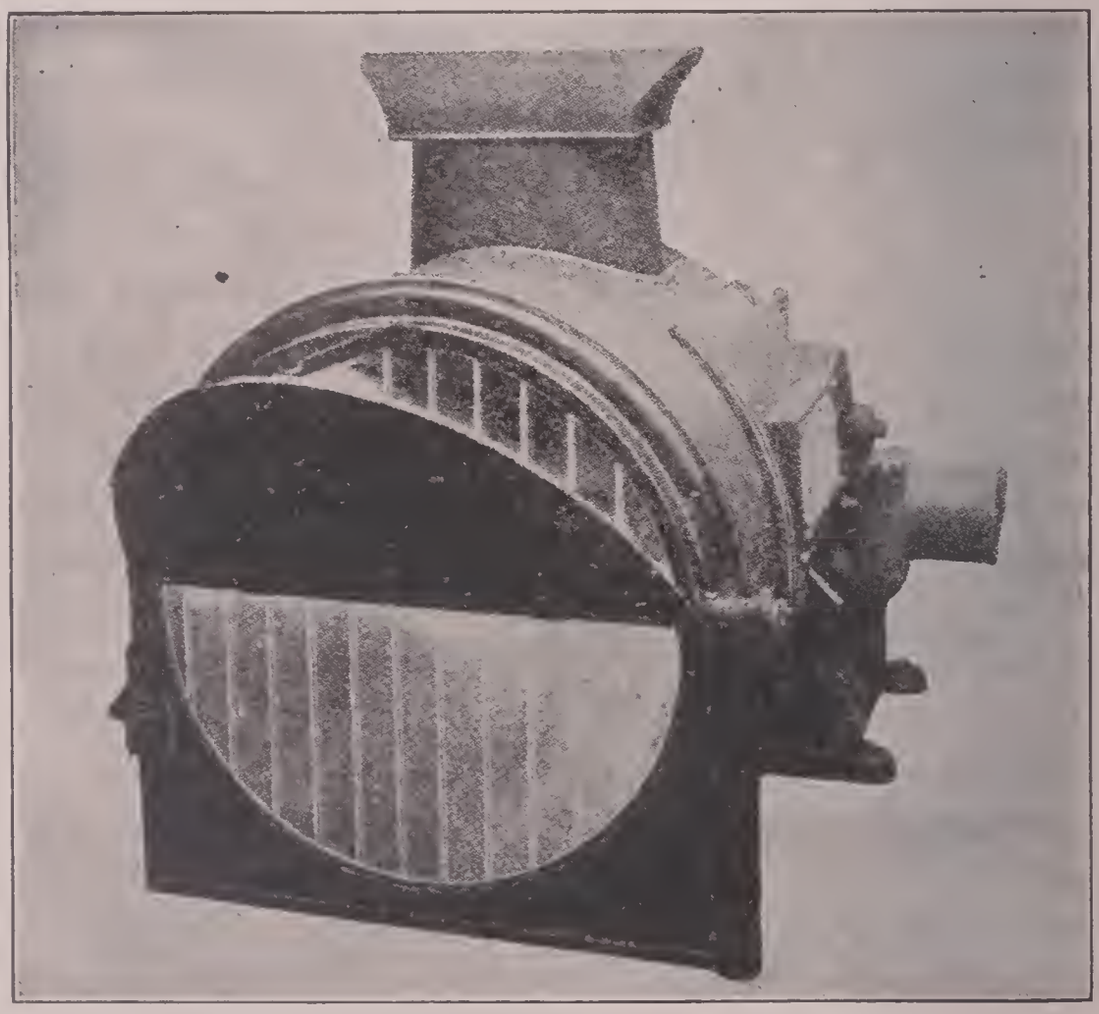


Fig. 189. Front View Showing Shade Drawn.

The Edwards equipment, a front view of which is given in Fig. 189, consists of four parts: first, the motor, a simple-acting steam turbine; secondly, the dynamo,

mounted on the same axle with the turbine and designed to yield to the arc light a current of from 30 to 33 amperes and from 30 to 33 volts; thirdly, the lamp, including the arc, the deflectors, and the case; and fourthly, the bed-plate on which the whole apparatus is mounted.

The steam turbine, shown disassembled in Fig. 190, is provided with a propellor wheel, which is wholly constructed of rolled steel. It has a factor of safety of about 7, for, while the normal speed of the engine and dynamo is about 2,000 R. P. M., the wheel will withstand successfully a speed of about 14,000 R. P. M. The speed of the engine is held constant, or practically so, regardless of change of load or initial pressure, by a simple and efficient governor, which is so arranged with relation to the other parts of the engine as to be easily and readily accessible, should occasion demand. The wheel shaft is journaled in ball bearings, and the coefficient of friction is so low that the turbine will operate, running to its full speed, under a pressure so slight that a pointer upon a 180-pound steam gauge will not leave its stop, the gauge being connected between the governor valve and the nozzle. All the moving parts are encased in a cast-iron housing, so designed as to thoroughly protect it from the elements, dust, dirt, etc. The lubrication is automatic, and is provided by loose rings feeding the oil to the ball bearings from the oil wells.

The dynamo is of peculiar construction, designed for the particular purpose for which it is used. The field is differentially wound, and the electric circuits so arranged that a burned-out armature is impossible. Should a short circuit occur on any point of the circuit, the current is neutralized, and no matter how long the engine may run or the armature rotate, there will be no pro-

duction of current whatever until the short circuit is removed. As soon as this is done the dynamo performs its proper functions and operates as usual. The current densities throughout the whole machine are very low, so that a minimum heat effect is produced, regardless of extremes of temperature or other conditions which might affect the resistance of the machine. Low-resistance carbon brushes are used, and many months of constant wear

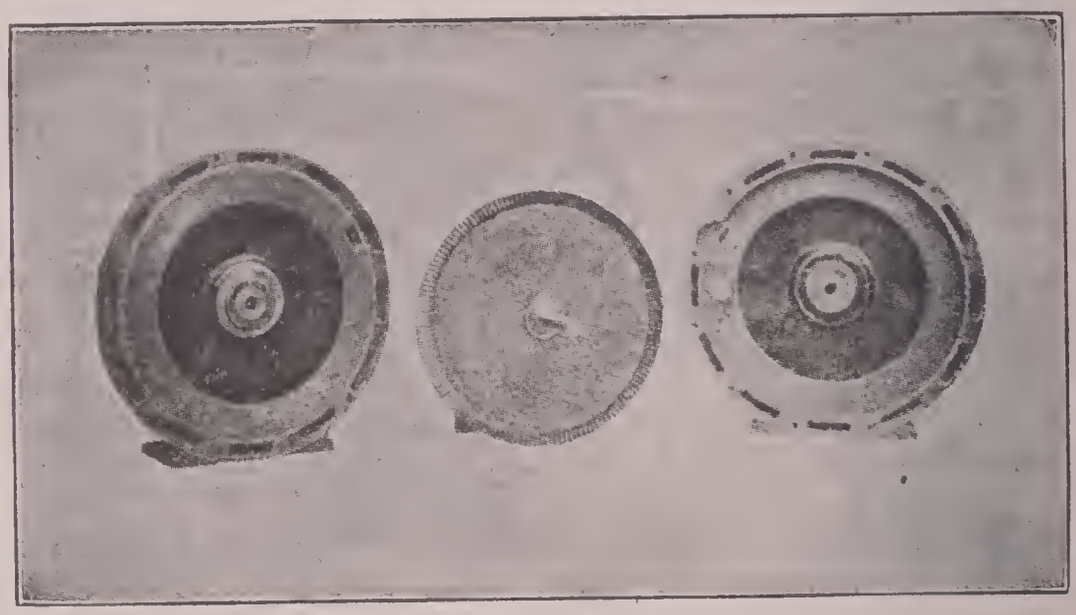


Fig. 190. The Turbine Engine Disassembled.

show very little deterioration of these brushes. Very large and long journal bearings are provided, and profuse lubrication is secured through the medium of loose rings dipping into the oil wells. An important feature of the equipment is the arc lamp with its parabolic reflector. It is strongly made, and care has been taken to insure a steady and constant light, free from flicker.

The vertical beam is caused to project upwards by an auxiliary plane deflector, placed outside the goggle at an angle of 45 deg. and in such a position as to intercept about 40 per cent of the whole volume of light issuing

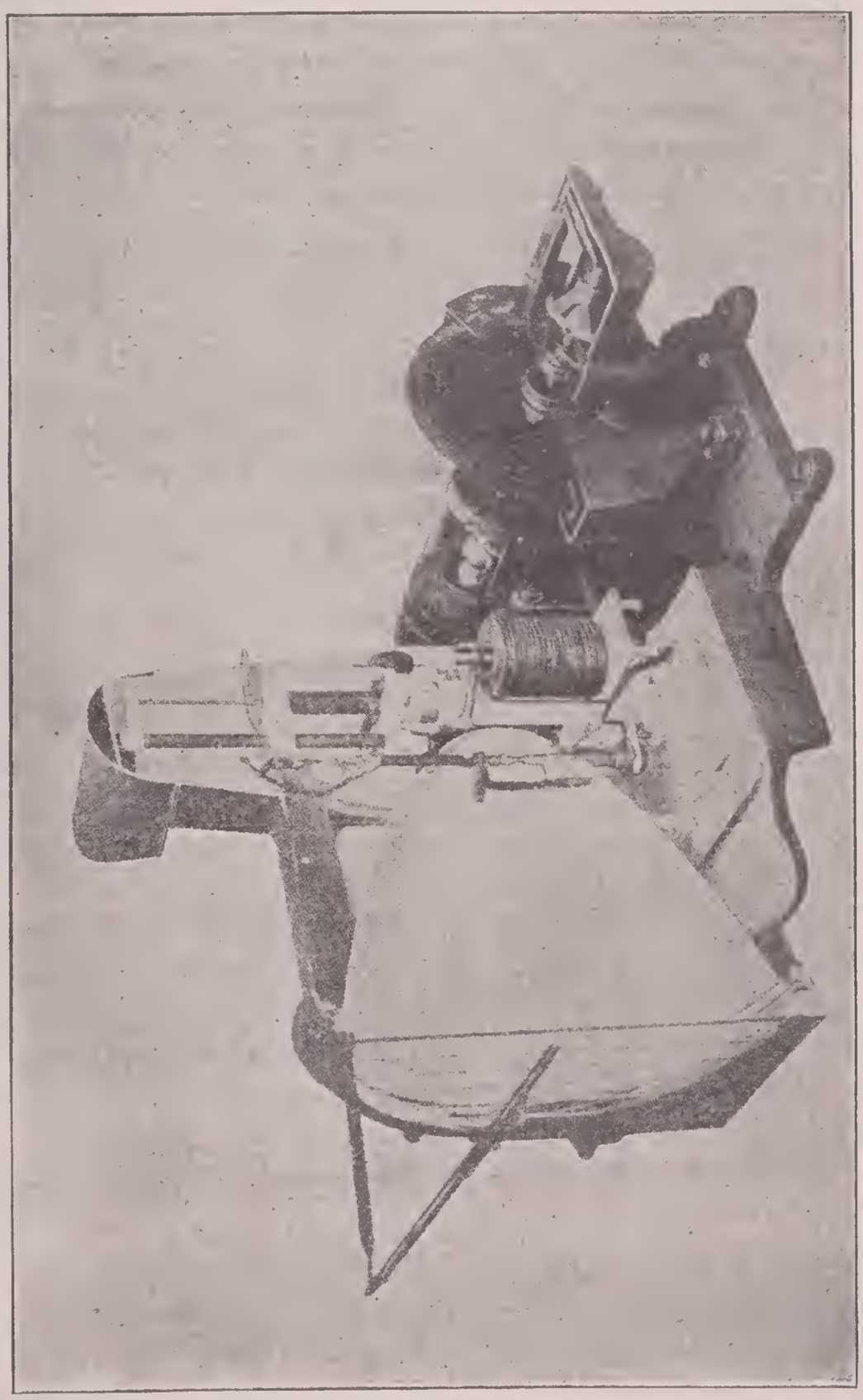


Fig. 191. A Sectional View of the Edwards Electric Headlight.

from the parabolic reflector, and direct it vertically. This vertical beam forms a constant warning signal. Reaching to a great height, and on cloudy nights striking the clouds, it can be seen for many miles. In fact, upon the Big Four road, it has been seen for a distance of 21 miles, and on the Chicago, Milwaukee and St. Paul road it has been seen for a distance exceeding 16 miles.

The whole apparatus is generally mounted upon one cast-iron bedplate, and it is the work of only six, or ten hours to apply the equipment to the locomotive. All that is necessary is to secure the bedplate at the proper place on the smoke arch by means of brackets bolted thereon, the running of a three-quarter-inch live steam pipe from the cab, and the passing of a 1¼-inch exhaust pipe into the smoke arch.

Figure 191 gives a sectional view of the equipment. The light is fitted with a carbon positive pole and a copper negative. The focus is maintained by special mechanism, an adjustable guide being furnished for the carbon. This guide bears in three points on the carbon and permits of perfect adjustment regardless of small differences in diameter of the latter. Fig. 192 presents a view of the assembled turbine and generator direct connected. The steam is led to the engine through a ¾-in. copper pipe from the cab, and the exhaust is led into the smoke arch by means of a 1¼-in. wrought iron pipe. The steam exhausts, almost opposite the admission port of the engine, through the various exhaust ports seen in the cut, Fig. 190.

To Remove the Armature. To remove the armature, first take out the brushes from their holders, back of the set screw on the brush holder yoke; then remove the four nuts and two cap screws. The end plate may then

be taken off, after which the armature may be withdrawn. The field coils are held in place by ¼-in. square keys. The coils may be removed by driving out the keys and removing the pole pieces. In reassembling the dynamo, be sure the oil rings are raised to permit the shaft to pass through the bearings, then replace the end plate. The field coils and armature are protected by a circular sheet steel casing fitting into grooves in the end plates. The armature is mounted in self-adjusting bronze bearings. These bearings are lubricated by means of rings, the shaft having spiral grooves to carry the oil throughout the bearings. Overflow oil holes are drilled in both oil wells to prevent too much oil being placed in the bearings. These overflow holes must be kept open, and the oil wells

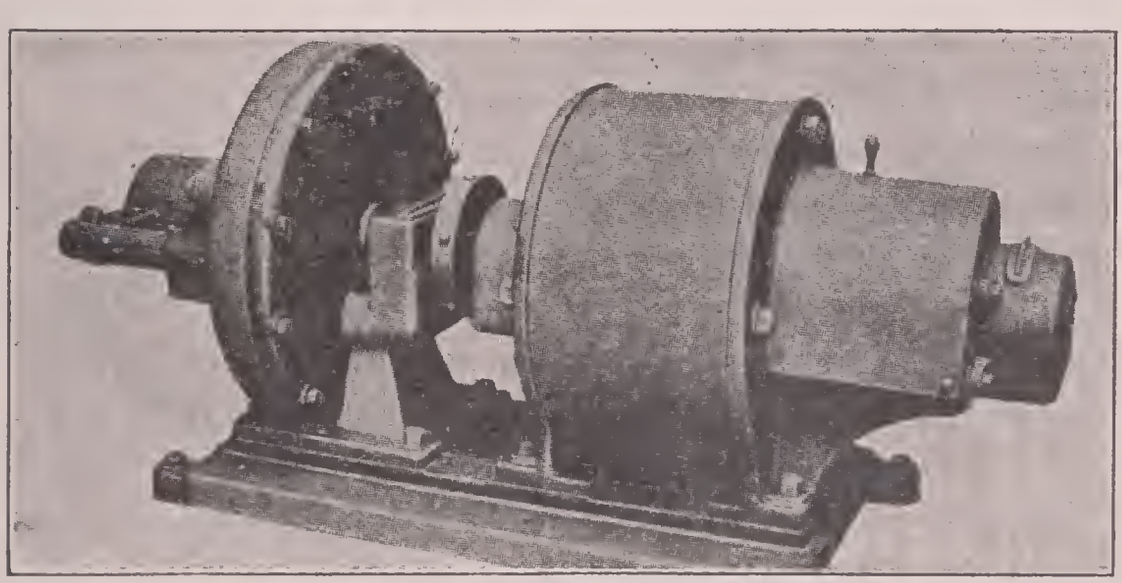


Fig. 192. The Miniature Turbine and Dynamo.

occasionally cleaned of sediment by removing the plugs, and cleaning the reservoirs with kerosene. Care must be taken to have the oil rings in the slots in the bronze bearings.

To Adjust the Lamp. To adjust the lamp, first have the correct speed on the dynamo, and the commutator

and brushes working properly, then set the adjusting screw on the shunt side of the lamp, so that the pawl will clear the escapement wheel about 1-100 of an inch; then raise the brass tube, or oil cylinder, which carries the carbon by means of the carbon holder support, and permit it to fall, then adjust the limit screw so that when the arc is established it will hold over without breaking the circuit. If the arc breaks when the light is started, or, if, while the light is in operation, the equalizer sets up a pumping action, giving a vibrating or flickering light, the limit screw is set too high. On the other hand, if this limit screw is set too low, there will not be sufficient separation, when the light is started, and there will be only a small red light. This also may occur when the light is in operation and the locomotive in motion, which defect may be easily and quickly corrected by slightly raising this limit screw. When the arc is properly established, the spring should be adjusted by means of the nuts, so that the carbon will feed as it burns away without breaking the arc. If "tack head" deposits form on the top of the negative, and, breaking off, interrupt the light occasionally it is evidence that the spring is too weak, thus not allowing the arc to be drawn out long enough. A slight increase of the spring tension by adjusting the nuts will correct this fault.

It is necessary that the brass tube, or oil cylinder, guide rod, and valve rod should be kept perfectly clean. For this purpose a soft felt cloth should be used, never using sandpaper or emery paper or waste, as the sand and emery will roughen these surfaces, and the lint from the waste may clog the rack.

The Brushes and Brush Holders. The brushes and

brush holders must be kept perfectly clean, and the brushes must always slide freely in the holders. The brushes must always occupy the proper position on the commutator, and the screw firmly set. Unless they are kept in this position there will be sparking between the brushes and the commutator, thus reducing the light and burning the commutator. The ends of the brushes which bear upon the commutator are slightly beveled, and in replacing them in the holders see that their full end surface has complete contact with the surface of the commutator. The brushes are self-lubricating and no oil or other lubricant must ever be put on the commutator.

Use only sufficient spring tension on the brushes to prevent sparking between the brushes and the commutator.

The Commutator. The copper bars of the commutator are separated by mica insulation.

The copper will wear more than the mica and, if the mica is allowed to project above the surface of the copper, even slightly, it will prevent perfect contact between the copper bars and brushes, which must, at all times, be maintained. To prevent the mica from thus interfering with perfect contact between the brushes and the copper bars, the mica should always be a little below the surface of the copper bars. This is accomplished by filing out the mica to a depth of about 1-64 of an inch by the use of a small file. This process will raise a slight burr on the edges of the copper bars which must be removed by using a strip of No. 0 sandpaper (never use emery). In this operation run the machine slowly, meantime working the sandpaper back and forth lengthwise of the commutator so as to cover the whole surface, until it is perfectly smooth, then wipe the commutator clean,

working a clean cloth, or waste lengthwise to clean out the shallow grooves between the copper bars.

Use sandpaper as above described whenever the commutator becomes slightly rough. Should the commutator become too rough, or out of round, the armature should be removed, and the commutator dressed off in a lathe, using a diamond-point tool, and removing only enough metal to make a perfectly true and clean surface. After turning, polish with sandpaper (never use emery), then file out the mica as above directed.

STRESSES ON PEDESTAL JAWS.

It is commonly supposed that the stresses upon each jaw are equal, but such is not the case.

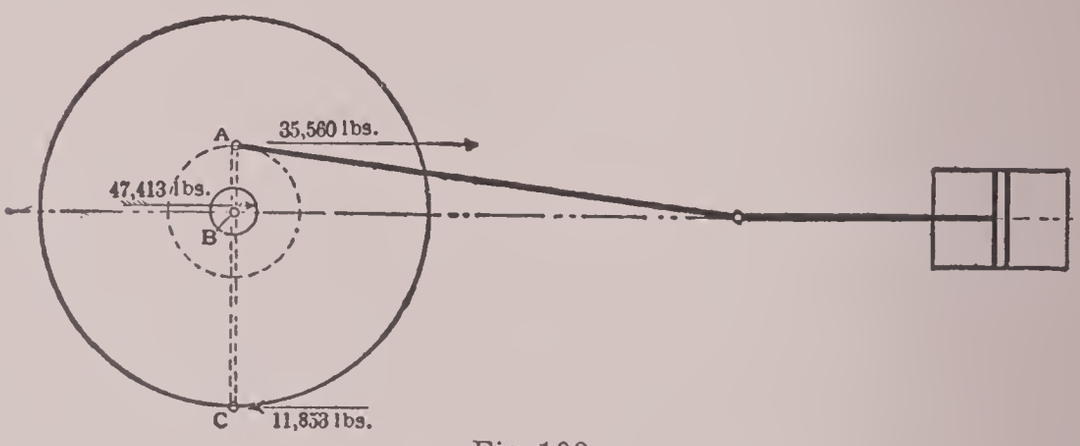


Fig. 193.

Suppose an engine having 18x24-inch cylinders, 72-inch drivers and carrying 175 pounds of steam. The engine is, by hypothesis, pulling a train, and we will further suppose that it is running slowly, so as to be in full gear. Then the maximum pressure on the pedestal jaws will be approximately, when the crank pins are on the upper and lower quarters. The area of an 18-inch diameter piston (neglecting the rod) is about 254 square

inches. With 175 pounds boiler pressure we may realize 140 pounds on the piston. Multiplying 254 by 140 gives 35,560 pounds, the thrust on the piston rod. Consider now this as a force, applied in a line parallel to the track, to the crank pin on the top quarter.

If the center of the crank pin and the point of contact of the wheel with the rail be considered the extremities of a lever A C (Fig. 193) 4 feet long, the fulcrum is at the rail C; the applicaton of the force to the pedestal jaw

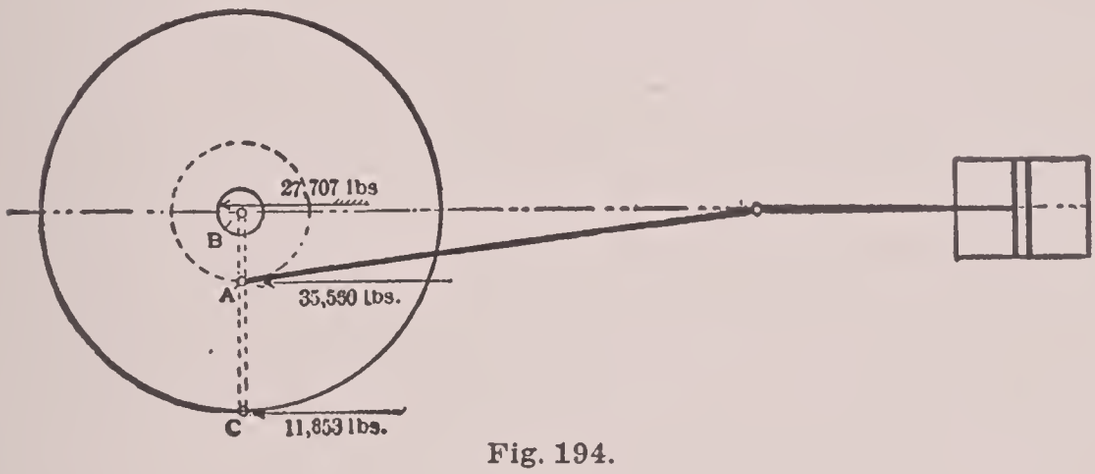


Fig. 194.

is at the axle B, 1 foot from the crank pin A, and the lever is one in the second order. The lever arms are here 3 to 4. Consider the forces in equilibrium and we have a stress of 35,560 pounds acting at the crank-pin end, 1 foot from the axle, or 4 feet from C. Multiplying 35,560 by 4, the length of the lever arm A C in feet, and dividing by 3, the length of the other lever arm B C, we obtain 47,413 as the number of pounds exerted at the axle against the front pedestal jaw.

The difference of these two forms, or 11,853 pounds, gives the force acting at the end of the lever, which in this case is resisted by the friction between the wheel and rail.

Consider now the fact that not only is the steam pushing the piston forward in the cylinders, but that it is also pushing on the back cylinder head with a force of 35,560 pounds. This is transferred by the frames, and as it is in an opposite direction to the force pushing forward on the front pedestal jaw, the difference of these two, or 11,853 pounds, must be taken as the net pull transferred to the draw-bar.

Let us now proceed to the other position, with the crank on the lower quarter. Here we have a lever B C, (Fig. 194) 3 feet long, one of whose ends B is the center of the axle, and the other the point of contact of the wheel with the rail C. The fulcrum is at the latter point as before, but the lever is now one of the third order.

With a force of 35,560 pounds on the crank pin A, as before, one-third of this will go to the rail and two-thirds to the axle, the forces being divided inversely proportionately to the respective lengths of the lever arms. This gives 11,853 pounds at the rail as before, and 23,707 pounds against the back pedestal jaw. Remember now, however, that the steam is also pushing against the front cylinder head with a force of 35,560 pounds, and this is transferred by the frames as before, so that the difference of these two amounts, or 11,853 pounds, is the net effort applied to the draw-bar.

We therefore see that the stresses are double as much on the front pedestal jaw as they are on the back one.

SHOES AND WEDGES.

Shoes and wedges are castings fitted to the pedestal jaws; they are also planed to fit into the grooves on the forward and back faces of the driving boxes and form a

bearing between the driving box and the pedestal jaws. The object of these castings is to hold the driving box in its correct position and at the same time permit of its adjustment when desired. The shoe is usually fitted to the forward jaw and the wedge to the back jaw; the wedge should be so adjusted as to take up all lost motion between the forward and rearward jaw, but not so tight as to cause the box to stick; in other words, prevent the driving box from moving up or down in the jaw. The two flanges on the driving box prevent it from working laterally. The importance of having the shoes and wedges properly fitted upon a locomotive cannot be overestimated, and yet there are railroad shops in which this kind of work is considered inferior, or second class work, while in reality there is no job upon the locomotive of more importance, as a set of shoes and wedges improperly fitted up may cause the engineer much trouble with rod brasses and driving box brasses, or cause the tires to cut, and sometimes break crank-pins or side-rods.

SETTING UP WEDGES.

An engineer should have steam up when the wedges are adjusted, because the parts of the frame that lie against the fire box become hot and expand, or get longer than when the boiler is cold. The crank pins should be on the top quarter when the wedges are set up; a block may be placed on the rail back of the wheel and the engine moved back against it, or the wheel may be pinched forward with a pinch bar, the object being to jam the driving box up solid against the shoes so that whatever lost motion there may be between the jaws will be at the back of the box, so the wedges will slip up freely. When

you have all the boxes up tight against the shoes, set the wedges up moderately tight with a 12-inch monkey wrench and tighten the jam nuts with a large wrench, because the pedestal braces, or binders, are tight before setting up the wedges. The reason the crank pins should be upward on the side that the wedges are being adjusted is because if the pins were on center and the side rods were too long or too short they would necessarily force one of the driving boxes against the wedge instead of the shoe; and if the crank pins were below the centers when the wheel was pinched forward the side rod would have a tendency to draw back on the crank pin and would therefore draw the box back against the wedge. While if the crank pins were on the top quarter, the tendency would be to force all the boxes on that side against the shoe. It would be well to slack all the rod keys before setting up the wedges if time will permit, and tram the wheel centers when you have finished. A very good way to set a wedge is to screw it up as far as possible, then at the top of the wedge, scribe a line on the frame, then draw it down about ⅛ of an inch and jam the nuts on the wedge bolt. If there is lost motion at the head of the wedge bolt you must make allowance for it. If there is another wedge bolt passing through the jaw tighten it and be sure that it does not extend through the wedge and rub on the driving box; measure its length.

THE PROPER TIME TO SET UP WEDGES AND KEY UP RODS.

After an engine has finished its run and is taken from the train every bearing is expanded as much as it will ever be under ordinary circumstances, and that is the time to key up main rod brasses and set up wedges.

CAUSE OF VALVES SOUNDING "OUT OF SQUARE."

Sometimes an engine's valves will sound "out of square" when the valve gear is perfectly adjusted. The following are some of the causes which may produce such an effect:

Driving wheels improperly quartered.

A main driving axle bent.

A patch inside the steam ports, or ports of different size.

Cylinders of different size (not compound engines).

Eccentrics of different throw.

Links of different radius.

A hole in the petticoat pipe or stack.

A leak at the exhaust pipe joint.

A valve-yoke cracked on one side.

Cylinders working loose on the frames.

POUNDING OF PISTON VALVES.

A matter in connection with piston valve locomotives which is of interest, and which has prompted some inquiry, is the pounding and consequent development of lost motion in the valve gear. This pounding is peculiar to the solid inside admission type of piston valve, and results from the pressure of the exhaust steam which, passing up over the ends of the valve exerts a force sufficiently great to take up the lost motion in the valve gear with a jerk or pound, more or less, according to how the engine is being worked. The exhaust pressure, it will be understood, operates to move the valve in the direction in which it is traveling at the time. The lost motion in the valve gear being taken up by the

effort of moving the valve, it is consequently free to be forced ahead the distance which the lost motion in the connections will allow. In the case of an engine considerably worn and being worked slow and in long cut off, this pounding is excessive and results in rapid wear and increase of lost motion. This action soon becomes severe enough to crystallize and break valve stems and other parts. It also causes a certain irregularity in the admission of steam, since the valve jumps and then stops, until the lost motion is taken up. This effect is perhaps not important, but the pounding is so detrimental to the gear as to probably preclude the use of solid piston valves, when the matter comes to be thoroughly understood.

KEYING UP RODS.

Side Rods. Owing to the general adoption of solid end rod brasses this operation will soon be a thing of the past. Solid end rods are much safer, and prevent many accidents with rod straps and broken bolts, and save many crank pins that would otherwise be broken on account of improperly keyed side rods, but as many locomotives are still in use which have strap rods with keys, we shall explain the proper method to set up rod keys. We will begin with the side or parallel rods. First select a level piece of straight track without any high or low joints, place the crank pins parallel with the wheel centers on one side, and beginning with the middle connection, if there is one, key the forward pins last on either eight, ten, or twelve-wheeled engines; drive the keys down moderately tight with a soft hammer, or hand hammer (not the coal pick), then key the other pins on that side the same way, then try the rods on the other

centers opposite to those on which they were keyed, and see that they will move laterally and do not clamp the pin; it is well, also, to try them on the quarters, as old pins may be worn out of true; now key the rods on the opposite side of the engine exactly the same way, leave your side rods free; it is no injury to have them rattle a little, especially if the driving boxes are well worn; when finished tighten all set screws.

Main Rod. When keying up a main rod place the crank pin on the *forward top eighth,* which is midway between the forward center and the top quarter. The reason for this is that the greatest wear on the pin takes place under the greatest pressure of steam; in other words, when the valve has reached its greatest opening. A brief study of the valve's motion will show you that steam is first admitted to the cylinder at or near each dead center, and as the average cut-off for all kinds of service is considered half stroke, it follows that steam will be cut off when the crank pin is at or near the quarter in each stroke; therefore when an engine is in the forward motion the wear on the pin will take place while the crank pin moves from the forward center to the lower quarter, and from the back center to the top quarter, and the valve must open and close with each stroke while the crank pin passes through this space. It therefore follows that the valve will have reached its greatest opening when the crank pin is at or near a point midway between the forward center and the bottom quarter and also between the back center and the top quarter. These points are called the lower front eighth and the top back eighth, respectively; the pin must therefore be the smallest at these two points, and as a main rod brass should always be keyed up on the largest part of the pin,

the pin should occupy a position directly opposite to the two points mentioned; therefore place the pin on the forward top eighth. Key the brass moderately tight, leaving it loose enough to allow of its being moved sideways on the pins, both front and back ends, then tighten the set screws. Some engines have so little clearance that it is safer to key the main rod with the pin on the dead center. Locomotives should have steam up when keying up the rods, as the frames are liable to expand enough to bind the brasses, if the keying is done while the engine is cold. It is very convenient also to have the steam to move the engine with. It is best to adjust the valves also while the engine is hot.

REPAIRING CYLINDER PACKING.

Engineers are not expected to do this kind of work and should only do it in case of emergency out on the road. If your packing blows report it without delay. When it becomes necessary to overhaul the cylinder packing, if steam is up, too much attention cannot be given the throttle while the cylinder-head is off, otherwise you may get a severe scalding.

First close the throttle and tighten the thumb screw, then open the cylinder cocks. Now move the engine until the piston-head is within one or two inches of its extreme forward travel. If you cannot move the engine then the main rod must be disconnected, but before doing this, scribe a line on the guides true with the back end of the crosshead and when recoupled again the crosshead should be true with the same line (the liners sometimes become misplaced). Also place a .wooden block between the guides in front of the crosshead; this

is merely for safety to prevent knocking out a front cylinder head. Now when you have the rod disconnected, move the crosshead forward within an inch or two of its extreme forward travel. By moving the reverse lever you can then cover both steam ports on that side to keep the steam out of the cylinder, then take off the cylinder-head and follower, make a mark on the cylinder-head; also on the follower before you take them down, so that you can put them up to the same places. (Sometimes the holes only fit one way.) Be careful to lay the cylinder-head nuts and follower bolts in such a manner when you remove them that you will be able to put them back where you found them. Now if solid ring packing, peen each packing ring inside until it fits the cylinder nicely; if it is Dunbar packing with springs, open out the springs until they will hold the packing securely against the cylinder. If it is a solid piston head, the piston and heads must be removed before you can remove the packing rings; when replacing the packing use a pair of inside calipers; hermaphrodites, or a pointed stick the right length will do; set the piston slightly above the center of the cylinder.

Be sure that the packing lies against the cylinder all around; if it does not, the packing should be taken out and the cause ascertained and remedied, if possible. After the packing is set out, clean off the follower and put it on. See that the heads of the follower-bolts press the follower, and that their ends do not touch bottom; then clean the cylinder-head joints, and put the head on. In screwing up follower-bolts, and cylinder-head nuts, be sure to get them solid. Use judgment especially in screwing the cylinder-head nuts, or you will break the studs. When the head is put on, see that the joints lie

close together all around; then put the top nut on and run it on until it just comes against the head; then put the bottom nut on in the same manner; then a nut on each side, and draw them, watching that the joints are together all around; then put on all the nuts and draw them equally all around the head. You will sometimes find, after removing the follower, that the packing is not slack, although it seemed so before the follower was taken off. This shows that the packing was clamped and held by the follower, and that the packing is too long—it was follower-bound. This can be remedied by placing a piece of wrapping paper between the follower and spider. Packing does not need setting out if it blows only a little at starting—you rob the engine of its power by having it too tight. It should not be snug enough to prevent the engine from drifting freely. Cylinder packing should not be allowed to run longer than two months, without being inspected.

WHAT AN ENGINEER SHOULD ATTEND TO BEFORE LEAVING AN ENGINE HOUSE.

When an engineer is required to take an engine out, he should try how much water is in the boiler, and see that the fire is in good condition. Examine every part of the engine and tender, to see that nothing is broken. Test the rod-keys; don't depend on set-screws; try to drive every key back with a soft hammer; examine every bolt and nut; test the brakes; see that they apply and release easily, especially if using automatic or air brakes. See that you have a good supply of fuel, and a tank full of water; all necessary tools, especially for firing, and that all lamps and signals are in good order and ready

for immediate use, and that there is sufficient oil and tallow and a box full of sand. You should also have a pinchbar and a pair of jacks in good order. If you have no jacks get four wedges of oak, about three feet long, four inches square at one end tapered to an edge one way at the other end. You should also have an axe and blocking for cross-heads, and a hand saw.

STARTING A TRAIN.

In starting a train, always have the reverse lever in extreme full throw, never cut back; after the train is well under way cut back one notch at a time; cut back as far as circumstances will permit; run with throttle wide open; you save steam better by cutting back with the reverse lever, if the notches in the reverse lever rack are close together, than by closing the throttle, and if drifting without steam, place the reverse lever at full stroke, allowing the valves to have full travel.

The reason that a train should not be started without having the reverse lever set at full stroke, is because there is a greater pressure on the valves when steam is first admitted to the steam chests than after the engine gets under headway; while the engine is running and using steam a thin film of steam is between the valves and their seats all the time, so the valve is not in actual contact with the seat, and if the valves are allowed to move over a portion of their seats without having steam (which of itself acts as a lubricant), or something to relieve the friction between them, the seats will in time become worn in the centers; for the same reason the lever should be placed at full stroke while the engine is drifting without steam to prevent, as much as possible, the

wearing of the valve seats in the center, and from wearing a rounding face on the valve.

The reason why an engine saves steam by running with the throttle wide open, and using the reverse lever to control the admission of steam to the cylinders, is that as the engine is cut back the lead on the valve increases and admits more pressure in the cylinder just where it is needed; that is, when the engine is at its weakest point right after passing its centers and nearly all of the full effective pressure of steam in the boiler remains in the cylinder nearly all of the time that the valve remains open, a little before the valve closes, the pressure entering the cylinder becomes what is known as wire drawn or weak steam. The same thing occurs if you run with your throttle partly closed, the steam becomes wire drawn before it reaches the steam chest.

If the throttle is wide open almost all of the full effective pressure of the boiler is always in the steam chest, and that pressure is at the valve ready to give its full impulse up to the piston, and is more effective than if the pressure was less, and longer continued by having the throttle partly closed, and having the valve travel further. If you use steam in a cylinder until the piston has traveled twelve inches, when the engine will do the work by the piston traveling but eight inches, you are wasting at least one-third of that extra steam, because you do not get the full effective pressure against the piston when it is most needed. If you strike a nail with a force of 120 pounds it is driven farther than if struck with a force of 100 pounds. So is a piston driven with more force and a greater distance with 120 pounds pressure than it is with 100 pounds pressure.

Another point is that after a piston has moved five or

six inches from the end of the cylinder, not near as much steam is required to move the engine for the next third of the half revolution of the wheels, because the leverage on the driving wheel increases as the crank pin leaves its center, until the pin is up or down, the leverage decreasing gradually until the pin reaches the other center, but before you can reach that point you have had the full benefit of all of the expansive force of 120 to 140 pounds of pressure, and it is well known that the higher the pressure the greater the expansive force of steam, and by closing the throttle, the motion of the piston takes the pressure out of the steam pipes faster than the throttle allows it to flow; immediately after passing the center you want the highest initial pressure in the cylinder that you can get there, and when the steam has done its work in that end of the cylinder you want to get rid of it as soon as possible, because we want to get steam into the other end of the cylinder, and do not want pressure on both sides of the piston at the same time. If the pressure could all be let out of one end of the cylinder as soon as steam enters at the other end, and if there was a free passage for the exhaust, after the steam is discharged from the cylinder, there would be no back pressure to overcome; but then we need some steam in that end of the cylinder for a cushion for the piston while passing the center. The best constructed valve motion gives the smallest amount of back pressure, allowing just enough compressed steam to form a cushion and to give the full boiler pressure against the piston as soon as the valve opens, and the outside lead that is gained by the valve being cut back, is also gained on the exhaust cavity in the valve, and thus you begin to release the steam sooner than if you use the throttle instead of the reverse lever.

Of course, there is more compression of steam between the cylinder head and piston at the end of the stroke, on account of increased lead, and the exhaust is somewhat choked, because the cavity of the valve does not open so wide, when the engine is cut back, but four or five inches of steam saved at each end of a cylinder more than makes up for all these losses. Your steam gauge will show that your engine will steam better with any train it can haul at maximum speed with valves cut back as far as circumstances will admit with throttle wide open, than with reverse lever two or three notches ahead with the throttle partly closed. The exceptions to this rule are very rare.

A WORD TO YOUNG RUNNERS.

One of the hardest things for a young runner to learn is the art of letting well enough alone. He usually monkeys with his wedges, rods, diaphragm, or petticoat pipe, or something that if left alone would answer a better purpose. He cannot be too careful about oiling, but this does not mean to wash the engine down with a half gallon of oil at the beginning of a trip and then letting her run until the babbitt begins to fly, but at every station when there is time enough to put a few drops on every wearing surface, and lay his hand on every such surface expecting to find it hot. Another thing he will find to be a great satisfaction to him is the thorough inspection of every part of the engine and tender before starting on a trip. It will give him more confidence while running that he is going to get through all right, if he satisfies himself that to the best of his knowledge the engine is in good condition before he starts, and if anything should go wrong he will not have himself to

censure for want of careful examination. Examine your engine and tender thoroughly before leaving the engine house, at the end of each trip and report all needed repairs. If there are any defects that can be remedied attend to them as soon as possible, no matter how trifling or unimportant they may seem.

TABLE OF TRAIN SPEEDS.

EXAMPLE.

If a train covers the distance between two mile posts in 65 seconds, the speed is 55.3 miles per hour.

TABLE 54.

Seconds Per Mile.	Miles Per Hour.	Seconds Per Mile.	Miles Per Hour.
40	90.	58	62.
41	87.8	59	61.
42	85.7	60	60.
43	83.7	61	59.
44	81.8	62	58.
45	80.	63	57.1
46	78.2	64	56.2
47	76.6	65	55.3
48	75.	66	54.5
49	73.4	67	53.7
50	72.	68	52.9
51	70.6	69	52.1
52	69.2	70	51.4
53	67.9	75	48.
54	66.6	80	45.
55	65.4	85	42.3
56	64.2	90	40.
57	63.1	100	36.

COMPUTING GRADES—THE DIFFERENT WAYS OF EXPRESSING.

Civil engineers, in computing grades, say, so much in 100, or such a per cent. Thus, a grade of three inches in 100 feet would be stated as "¼ in 100, or ¼ of one per cent." By the English method it would be called "one in four hundred." The American method always gives the grade per mile, which in this case would be "13 2/10 feet per mile."

Where the engineer said three per cent., or three in 100, the English would say, 1 in 33⅓, which, by American practice, means 158 4/10 feet per mile.

To reduce grades stated in per cents. or the rise in 100 feet, to feet of grade per mile, multiply by 52 8/10. Thus a three per cent grade reduced to feet per mile is expressed $3 \times 52\ 8\text{-}10 = 158\ 4\text{-}10$.

To reduce grades stated in the English method to feet per mile, divide 5,280, the feet in a mile, by the number stated; thus 1 in 25 would be expressed, $5{,}280 \div 25 = 211\ 2\text{-}10$ ft. per mile.

PRIMARY CONSIDERATIONS IN LOCOMOTIVE DESIGN.

The design of a locomotive engine presents certain primary points for consideration that are of the first importance, and are mutually related and inter-dependent. Upon the correctness of their determination depend the efficiency and economical performance of the work for which the engine is intended. They may be enumerated thus:—

1. *The Velocity,* or running speed.
2. *The Train Resistance,* dependent in part on velocity and due in part to gravity.

3. *The Tractive Effort,* or force of traction, which at least must be equal to the train resistance.

4. *The Force of Adhesion,* which is dependent upon the adhesive weight.

5. *The Wheel-base,* which to a great extent determines the adhesive weight.

6. *The Effective Steam Pressure* in the cylinders.

7. *The Heating Surface* of the boiler.

If the value of these quantities be not carefully arranged in due proportion to each other as the first step in the design of a locomotive, no amount of subsequent correctness in matters of detail will compensate for their inaccuracy.

The velocity or running speed is of course determined by the character of the service, whether freight or passenger.

Train Resistance may be divided into two parts:—(1) Resistance dependent on velocity; (2) resistance due to gravity. It is also considerably modified by (3) the effect of wind and curves.

(1) *Resistance dependent on Velocity* is due to the friction of the moving parts of the engine and vehicles and to the displacement of air at the given speed—say, V miles per hour. The determination of this component of resistance really depends upon experimental enquiry. De Pambour, in 1836, investigated the matter, and Gooch, in 1848, communicated to the Institute of Civil Engineers the results of a series of experiments on the subject. From these Mr. D. K. Clark deduced his well-known formula

$$R = 8 + \frac{V^2}{171}$$

where R = train resistance in pounds per ton for en-

gine and train combined, and involves the assumption that the weather is calm and the rolling stock and track in good condition. This formula has been modified for oil lubrication into

$$R = 6 + 0.009\ V^2$$

Harding, on the ground that one of the factors producing resistance to motion was the work done in moving the air from the front of the train, proposed formulæ involving the area of train frontage. Since this area, however, does not vary greatly in different kinds of rolling stock it is doubtful whether this is of any advantage.

Harding's rule:

$R = l\ (6 + .34\ V) + .0025\ V^2A$; or as an alternative
$= l\ (6 + .067\ V) + .00002\ V^2B$

where l = gross train load; A area of its frontage in sq. ft.; B its volume in cub. ft. Petigrew proposes the following formula,

$$R = 9 + .007\ V^2$$

By George R. Henderson (Proc. Engrs. Club of Phila., 1886):

$$R = 0.0015\ (1 + V^2 \div 650),$$

in which R = resistance in lbs. per ton of 2240 lbs. and v = speed in miles per hour.

Speed in miles per hour:

5	10	15	20	25	30	35	40	45	50	55	60

Resistance in pounds per ton of 2000 lbs.:

3.1	3.4	4.	4.8	5.8	7.1	8.6	10.2	12.1	14.3	16.8	19.2

By D. L. Barnes (*Eng. Mag.*), June, 1894:

Speed, miles per hour.....	50	60	70	80	90	100
Resistance, pounds per gross ton.	12	12.4	13.5	15	17	20

(2) *Resistance due to Gravity.*—In ascending an incline the train is raised vertically a distance depending upon the length and inclination of the grade. Thus let a triangle A B C represent an incline, and W the load to be hauled up it. Also let a perpendicular E F be made proportional to the load W, and lines E G and F G be drawn through E and F respectively at right angles to and parallel to A B. Then E G represents the reaction of the plane, and F G will be the tractive power required to neutralise the force of gravity, or, in other words, the resistance due to gravity. Also, since the triangles A B C and E F G are similar,

$\frac{F\,G}{F\,E} = \frac{A\,C}{A\,B} = r$, the rate (or sine of the angle) of inclination, and $F\ G = r \times E\ F$. But E F represents W; therefore the tractive power $= r\ W$.

For instance, let $r = \frac{1}{224}$, and $W = 1$ ton,—then $\frac{1 \times 2240}{224} = 10$ lbs. will be the resistance per ton due to gravity.

From the above we see that the sum of the resistances due to velocity and gravity will be, in pounds per ton of gross load,

$$R = 2240\ r + (9 + 0.007\ V^2) \quad . \quad . \quad . \quad .$$

To find the gross resistance for any train load l in tons, the above resistance R must be multiplied by l. When the train is ascending an up grade with inclination r at con-

stant speed, it should be noted that the work done in the cylinders per minute will be equal to R V, where v is the speed in feet per minute. If the speed is increasing, the work done in the cylinders will be greater than R v by the amount necessary to produce the acceleration, while for diminishing speed R v will be greater than the work done in the cylinders.

The following rule for the resistance of curves is given by Morrison:—

Let R = Radius of curve.
L = Length of rigid wheel-base.
W = Weight of vehicle.
D = Distance of rails apart from tread to tread.
F = Coefficient of friction of wheels on rail = according to the weather from 0.1 to 0.27.

$$\text{Resistance due to curve} = \frac{W\,F\,(D + L)}{2\,R}$$

The following five rules relating to resistance and tractive power have been adopted by the Baldwin Locomotive Works of Philadelphia, and have proven to be reliable.

SPEED RESISTANCE, LOCOMOTIVE AND TRAIN.

(Rule 1.)

$$R = 3 + \frac{V}{6}.$$

R = resistance, in pounds, per ton of 2000 pounds;
V = speed, in miles, per hour.

This formula represents the resistance for sustained speed, and the element of acceleration is not taken into

consideration. It is deduced from the results obtained by comparison of a large number of indicator cards taken at various speeds.

GRADE RESISTANCE.

(Rule 2.)

The resistance for a straight grade of 1 foot per mile is 0.3788 pound per ton.

If

G = grade, in feet, per mile;
T = weight of train, in tons (2000 pounds);
R = resistance, in pounds.

$$R = 0.3788\ GT.$$

CURVE RESISTANCE.

Taking the curve as expressed in degrees of deflection from a tangent measured from stations 100 feet apart, the resistance of curves may be expressed as in proportion to the number of degrees in the curve. The resistance naturally varies with the construction of the roadbed, speed of train, and other conditions of service, so that no general rule can be expected to apply to all cases. Approximately, with moderate speed and under ordinary conditions, the resistance may be computed on the basis that each degree of curvature is equal to a straight grade of 1½ feet per mile.

(Rule 3.)

The following formula corresponds to this allowance:

Let

A = angle of curve, in degrees;
T = weight of train, in tons;
R = resistance, in pounds.

$$R = 0.5682\ AT.$$

ACCELERATION RESISTANCE.

(Rule 4.)

The resistance opposed to the acceleration of a train from any speed to any higher speed may be computed by the following formula:

Let

R = the resistance, in pounds;
T = weight of train, in tons (2000 pounds);
V = initial speed, in miles, per hour;
V' = accelerated speed.

$$R = 0.0132\ (V'^2 - V^2)T.$$

Thus, for a weight of 1 ton and an acceleration from 30 miles per hour to 50 miles, we have

$$0.0132\ (50^2 - 30^2) =$$
$$0.0132\ (2500 - 900) =$$
$$0.0132 \times 1600 = 21 \text{ pounds,}$$

and this, multiplied by the weight of the train, in tons, will give the total resistance due to the acceleration.

TRACTIVE POWER.

(Rule 5.)

Let

d = diameter of cylinder;
l = length of stroke;
D = diameter of driving-wheels, in inches;
T = tractive power, in pounds, per pound of mean effective pressure in cylinder.

$$T = \frac{d^2 l}{D}$$

The mean effective pressure may be taken as equal to 85 per cent. of the boiler pressure.

The tractive power of a locomotive multiplied by the speed, in miles, per hour, divided by 375, gives the horse-power.

Compound Engines.—To calculate the tractive effort of compound locomotives, von Borries has given the following rule:—

If d = diameter of low-pressure cylinder.
T = tractive power.
D = diameter of driving-wheels.
p = boiler pressure.
s = stroke of piston.

Then $d^2 = \frac{4\,T \times D}{p \times s}$: whence T, the tractive effort, can be easily found.

The Baldwin Locomotive Works give the following formulæ for compound engines of the Vauclain four-cylinder type:

$$T = \frac{C^2S \times \frac{2}{3}P}{D} + \frac{c^2S \times \frac{1}{4}P}{D}$$

T = tractive power in lbs.
C = diam. of high-pressure cylinder in ins.
c = diam. of low-pressure cylinder in ins.
P = boiler-pressure in lbs.
S = stroke of piston in ins.
D = diam. of driving-wheels in ins.

For a two-cylinder or cross-compound engine it is only necessary to consider the high-pressure cylinder, allowing a sufficient decrease in boiler pressure to compensate for the necessary back-pressure. The formula is

$$T = \frac{C^2S \times \frac{2}{3}P}{D}$$

ADHESION.

The limit of the hauling capacity of the locomotive is the adhesion due to weight on drivers.

In English practice 0.15 of the load on drivers is allowed in dry weather, and 0.07 in damp weather. In American practice the adhesion is generally taken at from .25 to .20 of the load on the drivers.

Hauling capacity at slow speed on varying grades may be calculated by the following formula: Let $T =$ tons of 2000 lbs., locomotive and train, per 1000 lbs. load on drivers, $a =$ the reciprocal of the coefficient of adhesion, $g =$ the per cent of grade, $R =$ the frictional resistance in lbs. per ton. Then $T = \dfrac{1000 \div a}{R + 20g}$

From this formula the following table has been calculated:

Grade Per Cent,	0	0.5	1	1.5	2	2.5	3	3.5	4	5	6	7
Tons Hauling Capacity per 1000 lbs. Weight on Drivers.												
For $a = 4$, $R = 6$.	42	15.6	9.2	6.9	5.4	4.5	3.8	3.3	2.9	2.4	2.0	1.7
$a = 5$, $R = 8$.	25	11.1	7.2	5.3	4.2	3.4	2.9	2.6	2.3	1.9	1.6	1.4
$a = 5$, $R = 10$.	20	10.	6.7	5.	4.	3.3	2.9	2.5	2.2	1.8	1.5	1.3

TRACTIVE POWER OF A LOCOMOTIVE.—*Single Expansion.*

Let $P =$ tractive power in lbs.

$p =$ average effective pressure in cylinder in lbs. per sq. in.

$S =$ stroke of piston in inches.

$d =$ diameter of cylinders in inches.

$D =$ diameter of driving-wheels in inches. Then

$$P = \frac{4\pi d^2 pS}{4\pi D} = \frac{d^2 pS}{D}$$

The tractive power of Vauclain four-cylinder compound locomotives may be ascertained by the following formula:

$$\frac{C^2 \times S \times \frac{2}{3} P}{D} + \frac{c^2 \times S \times \frac{1}{4} P}{D} = T, \text{ in which}$$

C = Diameter of high-pressure cylinder in inches.
c = Diameter of low-pressure cylinder in inches.
S = Stroke of piston in inches.
P = Boiler pressure in pounds.
D = Diameter of driving wheels in inches.
T = Tractive power.

TRACTIVE FORCE OF LOCOMOTIVES.—ADAMS.

d = diameter of piston in inches.
a = area of piston in sq. inches.
l = length of stroke in feet.
n = number of cylinders.
D = diameter driving wheel in feet.

Then the tractive force at circumference of driving wheels for each lb. per sq. inch mean effective pressure on piston,

$$= \frac{2\,a\,n\,l}{\pi D}, \text{ or } = \frac{d^2\,l}{D}.$$

Also let μ = adhesion of wheels to rails (say .2),
W = weight on driving wheels,
then $W\mu$ = maximum possible tractive force.

The greatest mean effective pressure on piston is commonly assumed to be 85 per cent of boiler pressure, but this will be different for each design of valve gear, other things being equal. The ordinary mean effective pressure on piston would probably not exceed 50 per cent of

boiler pressure. The tractive power of a locomotive decreases as the speed increases.

T = traction in lbs. for two cylinders.
p = boiler pressure lbs. per sq. inch.
d = diameter of piston in inches.
l = stroke in inches.
D = diameter driving wheel in inches.
K = coefficient = .65 for cut-off at ¾ stroke.

$$T = K\frac{p\,d^2\,l}{D}$$

—*De Pambour.*

ADHESION OF LOCOMOTIVE WHEELS.—ADAMS.

Locomotive driving wheels will commence to slip if the force at circumference equals about

1-5 of the load	= 448	lbs. per ton.
Westinghouse and Galton.	= 246.4	"
Poirée	= 465.9	"
Pennsylvania Railroad	= 550	"
Northern Pacific Railroad	= 670	"

—*Eng. Mech.*

Adhesion depends principally upon the state of the weather, and varies from a maximum of ¼ load to a minimum of 1-10 load, average say 1-6 load.

RESISTANCE ON RAILWAYS.—ADAMS.

Straight and level railway, in good condition, resistance (R) in lbs. per ton of total load (W)

$$= \frac{v \text{ miles per hour}^2}{171} + 8.$$

Do. on incline of 1 in $m = R + \left(\frac{1}{m}\ W \times 2240\right)$

On Prussian railways, R is taken at 1-100 W = 22.4 lbs. per ton.

By experiment in railway goods stations, R = 30 lbs. per ton moving slowly.

A train of 300 tons total can be hauled 40 miles per hour on a level with 600 I.H.P. (*F. W. Dean*). This gives a resistance of 18.75 lbs. per ton, assuming E.H.P. = I.H.P.

RAILWAY CURVES.—ADAMS.

W = maximum rigid wheel base of rolling stock in feet.
G = gauge of railway, i. e., inside measurement between rails in feet.
R = minimum radius of curve in feet.

$$R = 9\ W\ G.$$

DIMENSIONS OF CYLINDERS.

According to recommendation of a committee of the American Railway Master Mechanics' Association, the tractive force due to adhesion is taken as 0.25 of the weight on drivers for passenger engines, 0.24 for freight and 0.22 for switching engines, and the M. E. P. in the cylinder when developing the maximum tractive force is taken as 0.85 of boiler pressure, then let W equal weight on drivers, P equal tractive force in lbs., d equal diameter of cylinder, S equal length of stroke, and D equal diameter of drivers, all in inches.

$$\text{Then, } W = 4P =, \frac{4\,d^2\,p\,S}{D} = \frac{4\,d^2 \times 0.85\,p\,S}{D}$$

$$\text{Therefore, } d = 0.5\sqrt{\frac{DW}{p\,S}} = 0.542\sqrt{\frac{DW}{p\,S}}$$

Von Borries' rule for the diameter of the low-pressure cylinder of a compound locomotive is $d^2 = \frac{2ZD}{ph}$

where d = diameter of l.p. cylinder in inches;
D = diameter of driving-wheel in inches;
p = mean effective pressure per sq. in., after deducting internal machine friction;
h = stroke of piston in inches;
Z = tractive force required, usually 0.14 to 0.16 of the adhesion.

The value of p depends on the relative volume of the two cylinders, and from indicator experiments may be taken as follows:

Class of Engine.	Ratio of Cylinder Volumes.	p in percentage of Boiler-pressure.	p for Boiler-pressure of 176 lbs.
Large-tender eng's	1 : 2 or 1 : 2.05	42	74
Tank-engines ...	1 : 2 or 1 : 2.2	40	71

Horse-power of a Locomotive.—For each cylinder the horse-power is H.P. $= pLaN \div 33{,}000$, in which p = mean effective pressure, L = stroke in feet, a = area of cylinder $= \frac{1}{4}\pi d^2$, N = number of single strokes per minute, LN = piston speed, ft. per min. Let M = speed of train in miles per hour, S = length of stroke in inches, and D = diameter of driving-wheel in inches. Then $LN = M \times 88 \times 2S \div \pi D$. Whence for the two cylinders the horse-power is

$$\frac{2 \times p \times \frac{1}{4}\pi d^2 \times 176S \times M}{\pi D \times 33{,}000} = \frac{pd^2SM}{375D}$$

Prof. Henry Adams of London Engineering, gives the following simple formula for ascertaining the H. P. of a locomotive:

H.P. of locomotive = tractive force lbs. $\times$ V miles per hour $\times \frac{5280}{60 \times 33{,}000} = \frac{88\,T\,V}{33{,}000}$

TABLE OF PRINCIPAL ALLOYS.

A combination of zinc and copper makes bell metal.

A combination of copper and tin makes bronze metal.

A combination of antimony, tin, copper and bismuth makes britannia metal.

A combination of copper and tin makes cannon metal.

A combination of copper and zinc makes Dutch gold.

A combination of copper, nickel and zinc, with sometimes a little iron and tin, makes German silver.

A combination of gold and copper makes standard gold.

A combination of gold, copper and silver makes old standard gold.

A combination of tin and copper makes gun metal.

A combination of copper and zinc makes mosaic gold.

A combination of tin and lead makes pewter.

A combination of lead and a little arsenic makes sheet metal.

A combination of silver and copper makes standard silver.

A combination of tin and lead makes solder.

A combination of lead and antimony makes type metal.

A combination of copper and arsenic makes white copper.

STRENGTH OF VARIOUS METALS.

The breaking strain on various metals is shown in the following table, the size of the rod tested being in each case one inch square, and the number of pounds the actual breaking strains.

	Pounds.
Hard steel	150,000
Soft steel	120,000
Best Swedish iron	84,000
Ordinary bar iron	70,000
Silver	41,000
Copper	35,000
Gold	22,000
Tin	5,500
Zinc	2,600
Lead	860

To harden copper, melt together, and stir till thoroughly incorporated, copper and from one to six per cent of manganese oxide. The other ingredients for bronze and other alloys may then be added. The copper becomes homogeneous, harder and tougher.

A workman in the Carson mint has discovered that drill points, heated to a cherry-red and tempered by being driven into a bar of lead, will bore through the hardest steel or plate glass without perceptibly blunting.

TO BLACKEN BRASS.

Into 6 gallons muriatic acid put 2 pounds yellow arsenic and 1 pound black oxide of iron. Have the pieces thoroughly clean and dry and dip them until the desired shade is produced; then rinse them in clear water and dry in fine sawdust. A much smaller quantity of the chemicals may of course be used if the approximate proportions are secured.

Another recipe is: Make a strong solution of nitrate of silver in one vessel and nitrate of copper in another.

Mix the two together and plunge in the brass. The brass after being removed from the bath should be heated evenly until the required degree of dead blackness is obtained. This method is of French origin and was kept a secret for years, being used for the ornamenting of optical and other scientific instruments.

Another recipe is the following solution, recommended for obtaining a deep black color on brass: Dissolve 100 parts of copper nitrate in 100 parts of water. If the article is large, then paint it with this solution; if small, immerse it in the solution. After draining off or shaking off the excess of the solution, heat the article until the copper salt is decomposed into a black copper oxide. The work may be heated over a clear coal fire, but better results may be obtained by heating the articles in a closed muffle furnace. For some kind of work you may obtain more uniform results by using a weaker solution than given above, namely, 100 parts of copper nitrate to 300 parts of water.

Another recipe is the following solution recommended for obtaining a deep black color on copper and its alloys: Copper nitrate, 100 parts; water, 300 parts. The article, if large, is painted with it; if small, the article may be immersed in the solution. It is then heated over a clear coal fire and lightly rubbed. The article is next placed in or painted with a solution of the following composition: Potassium sulphide, 10 parts; water, 100 parts; hydrochloric acid, 5 parts. This composition need not be used in all cases. Small work can be much more conveniently treated by immersion in the solution first given, and after straining off or shaking off the excess of the solution, to heat the work on a hot plate until the copper salt is decomposed into a black copper oxide. It would be diffi-

cult to heat large articles upon a hot plate; a closed muffle furnace should give better results than an open fire. In any case, the heating process should not be continued longer than is necessary to produce the change mentioned above.

A POSSIBLE CAUSE FOR WRECKS.

By H. M. Perry.

(In American Master Mechanic, June, 1907.)

The investigations, both by the railroad and state authorities, as to the probable cause of the disastrous railroad wrecks of recent date, have brought out a number of theories, but we fail to find among them a word concerning a very prolific source of trouble in almost all railroad yards, namely, the derailment of cars and engines due to excessive loads on side bearings.

Many roads have had so much trouble of this kind with their bearings on locomotive tanks, while backing on curves, that they have stopped using them for switching except where it was absolutely necessary.

On a certain class of cars, the construction of which was such that the bottom frames were almost rigid, to such an extent that a jack placed under one corner would raise the end almost level, it was found that the only way of keeping them on the track on sharp curves was, to give from three-quarters to one inch clearance between the side bearings, with the result that it is almost impossible to ride on them, when running, owing to their excessive rolling.

The ordinary practice in passenger car construction and operation is to give only just a clearance between the side bearings, to prevent rolling, and as a result any

deflection in the bolsters brings the load down on the side bearings, which is the case in about nine-tenths of all cars now running.

As the outside rail on a curve is elevated from one-half to one inch for each degree of curvature and the easement or incline from the level rail to the elevation is from twenty to forty feet for each inch of elevation, it is reasonable to assume that as a car strikes a curve a large proportion of the weight of one end of the car is transferred to the outside bearings of the forward truck, due to the elevation of the rail, and at the same time centrifugal force comes into action to still further increase the load on this bearing.

As the load has to be overcome by the flange of the outside forward wheel pressing against the rail it can be readily seen that the factor of safety at this point, under present conditions, is not large.

Regarding the question of friction between side bearings, we find that standard authorities claim a factor of friction between cast iron surfaces of from 50 per cent for rough castings to 20 per cent for smooth castings, and if a wreck occurred with trains of comparatively new cars it is reasonable to assume that the side bearings would be rough castings, having a factor of friction of 50 per cent.

Under the foregoing conditions, we will assume that a sixty-five foot coach weighs 86,000 pounds, 60,000 pounds for the body and 26,000 pounds for the trucks, and carrying a load of 10,000 pounds; running at sixty miles an hour and strikes an eight degree curve; elevation of the outer rail 4 inches and center of gravity of the car body 6 feet 6 inches above the rail, the load equally resting on center plates and side bearings when car is on a tangent.

The weight of the body and load on the forward truck would be 35,000 pounds, of which something like 30,000 pounds would be on the outside bearing, due to the elevation of the outer rail and centrifugal force, and if these side bearings were of rough cast iron, as they would be under new cars, it would require a force of 15,000 pounds to slide one on the other. It would make very little difference if there was any clearance between the side bearings, when the car was on a level track, as the centrifugal force acting in a horizontal line through the car body would force it down on the outside bearing with a power about equal to one-half of the weight of the whole end of the car and load, while a large amount of clearance between the side bearings becomes positively dangerous, due to the rolling of the car and the added momentum of the rolling load.

Now in order to overcome this friction between the side bearings we have the pressure flange of the outside forward wheel against the rail, which on the car specified would amount to 48,000 pounds, the total weight of one-half of the body and load and one truck distributed over four wheels, or 12,000 pounds per wheel. Now with trucks having a seven-foot base and side bearings set thirty inches from the center plate, we have a leverage, from the flange of the forward wheel on the rail, to the center plate, of about fifty inches, against one from the center plate to the side bearings of thirty inches, or one of five to three, which would increase the force against the rail to about 20,000 pounds, plus the additional weight due to centrifugal force.

In none of the calculations have we taken into account several factors, all of which tend to increase the resistance of the truck in curving, or to reduce the available power

at the flange of the wheel against the rail. Additional resistance at the center plates, or inequalities in the castings of center plates or side bearings, rigidity in the under frames of cars which prevent them from winding, or twisting to allow adjusting themselves to the elevation of the rail on curves would both require more power to overcome. The axles in a truck being parallel instead of radial, together with centrifugal force, tend to carry a car in a straight line or to ride over the outside rail, or an increase in speed to the point where centrifugal force would counter-balance our small margin of safety would result naturally in a derailment.

And what is a remedy for these conditions? If ordinary rigid side bearings require from twenty to fifty per cent to overcome the friction between them, why not apply a roller side bearing that will reduce this friction to one or two per cent and reduce the danger of derailment in a corresponding degree? No one disputes the fact that the friction can be reduced in this way.

Aside from the question of safety, the economy of the roller side bearing has been demonstrated, and the facts show that they will pay for themselves in less time than the life of one set of wheels, in the saving of flange wear alone, to say nothing of rail wear or draft of train. A recent test of roller bearings on electric cars, showed a saving of 17.8 per cent in power required to haul the train.

In the face of these facts railroad men can no longer afford to neglect this matter and refuse to apply a remedy, when several years' experience have demonstrated that such a remedy exists and has been in practical use for a number of years.

INDEX

LOCOMOTIVE RUNNING AND MANAGEMENT

A

Absolute pressure 267
Absolute zero 270
Adiabatic curve 271-301-303
Adhesion 452-454
Admission 279-280
    inside 99-110
    outside 99-110
Air—quantity required for combustion.....37-38
Allen-Richardson balanced valve.....128-132
    advantages possessed by.....131-132
    defects remedied by balancing.....130
    dimensions of ports and bridges.....130
    distribution of steam by.....128-130
    object of designing.....128
Alfree system of steam distribution.....181-192
    advantages claimed for.....183-185
    arrangement of parts.....188-189
    compression controlling feature.....181
    form of main valve.....189
    form of compression valve.....189
    steam distribution by.....189-192
    outside admission 187
American balance valve.....165-176
    advantages claimed for.....166-167
    cylinder relief assured.....174-175

American balance valve—Continued.
formulæ for balance.....175-176
large area of balance.....167
rings and cones.....168-169-174
rules for applying 170-172
American semi-plug piston valve.....112-116
distribution of steam by.....113
internal admission 112
packing rings of.....112-114
American compound locomotive.....352-365
arrangement of cylinders.....352-353
by pass valves.....355
operation 355-357
Angular advance.....278
Austrian four-cylinder balanced compound.....388-390

B

Back pressure 267
Baker-Pilliod valve gear.....94-102
constant lead 101
outside admission 95
points of cut-off.....97-98
source of motion for.....94
weight of 99-102
Balanced compound 345-351
American Locomotive Co's.....349-351
Baldwin 347-349
distribution of steam.....348-349
efficiency in use of steam.....349-351
Balanced valves 116-192
Jack Wilson valve.....117-126
piston valve 108-115

Balanced valves—Continued.
Richardson 126-128
semi-plug 112-116
Balanced slide valves 148-180
Alfree system 181-192
Barnes valve 176-177
Briggs valve 179-180
Farrer's valve 146-147
Goodspeed valve 132-138
Haley's valve 138-142
Margo valve 180
Metzger's valve 142-143
Baldwin cross compound 368-369
distribution of steam in 369-373
intercepting valve 368-369
tandem compound 365-368
steam distribution 367
Boiler pressure 267
Boyle's law 271
Brass—how to blacken 458

C

Calorific intensity 16-17-18
power 18
values and flame temperatures 21
Cause of valves sounding out of square 433
Charcoal—composition of 27
how made 28
Clearance 270
meaning of term 50
mechanical 50
piston 270

Clearance—Continued.
purpose of 47
theoretical 294
valve clearance 47-48
Coal—composition 20-21
heating power of American.....22
heating power of British.....23
Coke—analysis of 25
composition of 20-21-24
furnace 24
gas coke 24
how prepared 24
Combustibles—chemical composition of.....19
heat value of.....20
Combustion 37-38
Compound engine 40
Composition of coal 20-22
coke 20-22
fuels 15
water 2-6
Compressibility of water 12
Compound locomotives 315-400
advantages of using 315-316
balanced 345-351
four cyl. bal. for Austrian Ry.....388-393
four cyl. bal. for Hungarian Ry.....390-395
French compound freight.....395-400
proper balancing of parts.....347
tandem 351-365
types of 316-317
Vauclain compound 318-345
Compression 48-270-271-279-280
Computing grades 444

Condenser pressure 268
Curve—adiabatic 271-301-303
expansion 271-295-298
isothermal 271-298
Cylinder—condensation 277
head broken 250
packing (repairing) 436-438

D

Diagrams—decrease of valve travel.....204-206
decrease of angular advance.....203-204
indicator 279-312
inside lap 207-208
point of admission or lead.....202-204
point of cut-off.....202-206-207
position of crank at cut-off.....202
Releaux and Zeuner—combined.....209-213
release and compression.....202-208-209
sample 279
showing one-half travel of value.....198
Dimensions of cylinders 455
Dunbar's packing 437

E

Eccentric—definition of term.....277-278
angular advance 278
position 278
rod broken 244-246
rod—correcting length of.....242

Edwards' electric head light.....419-428
adjusting lamp 425-426
brushes and holders 426-427
commutator 427-428
dynamo 421
removing armature 424-425
steam turbine 421-422
translucent shade 419-420
vertical beam of light 419-424
Electric head light.....405-406
Edwards' 419-428
Pyle's 406-413
Engine—compound 40
efficiency 273
ideal reciprocating 345-347
simple 40
Expansion—curve 271-295
ratio of 268
of water by heat.....2
of water by cold.....2
Explosive power of dust.....30

F

Farrer's slide valve.....146-147
advantages claimed for.....147
relief ports 146
Force—what it is.....272
Four cyl. bal. compound for Austrian Ry.....388-390
Hungarian Ry.....390-395
construction of fire-box.....391-394
location of cranks 391
steam distribution 392-393
weight in tons 391

French compound freight locomotive.....395-400
boiler dimensions 398-400
distribution of the steam 396-398
general dimensions 396
Fuels—classification of 20-21
calorific intensity 16-17-18
composition of 15
heat developed by.....16-39
heating power of.....18
gaseous 30
pulverized form 28-29

G

Gas—blast furnace 35
coal 30
flow of in pipes.....36-37
for gas engine.....35-36
natural 33
producer 31-33
process of making 30-31
Gaseous fuels 30
calorific power of.....32
Goodspeed valve 132-138
designed only for link motion.....137
longitudinal section of.....134
novel features of 135-136
plan view of 133
Gould balanced valve.....149-160
cover of chest 158
ease of operation 160
forms of 150-154-157
methods of balancing 150-152
packing bars 152-156-158

Gould balanced valve—Continued.
prevents excessive compression 152-153
principles of 150
springs 159
vacuum relief 159

H

Haley's slide valve 138-142
form of construction 139-140
independent face of 138-139
operation of 141-142
Heat—mechanical equivalent of 16
unit 16
Heating power of fuels 18-19-20
Horse power (H. P.) 270-286
calculations of 285-287
constant 274
indicated (I. H. P.) 270
net 270
of a locomotive 456

I

Ice—compressive strength of 4
expansive force of 4
of sea water 6
specific gravity of 4
specific heat of 4
weight of cubic foot of 4
Indicator, the 256-266
American Thompson 261
atmospheric line 258

Indicator, the—Continued.
care of 264
method of attaching 262
principles of 256-258
reducing mechanism 261-263
taking diagrams 258-259-264-266
tabor (the) 259-260
three way cock 262
Incompressibility of water 12
Initial pressure 267
Inspections to be made before leaving the round house 438-439
Isothermal curve 271

J

Jack Wilson valve 117-126
balanced area of 121
distribution of steam 122-126
faces of 117
pressure plate 120-122
sectional views of 119-123-125
Joy's valve gear 103-106

K

Keying up main, and side rods 432-434-436

L

Lap—purpose of 46-278
inside 46-278
outside 46-50-278

Lead—purpose of 50-278-279
adjustments for 54
cause of increase as cut-off is shortened.....
..... 50-53-54-214
Link—radius of 53
Locomotives— types of 40
cylinder and saddle castings 40-43

M

Mallet compound locomotive.....63-66
Maximum theoretical duty of steam.....272
Mean effective pressure (M. E. P.).....267
calculated by ordinates 304-310
methods of ascertaining 280-303-312
obtained with planimeter.....310-312
Momentum 272
Motion—first law of 271

O

Ordinates 277
methods of drawing 304-307

P

Petroleum oil—calorific value of.....21-22
heating power of 20
Peat—composition of 29
specific gravity of 29
Piston—clearance 270
displacement 270
speed 270

Piston valve 108-115
balanced 109-110
development of 160-161
fitted with packing rings 110-111
hollow 110-111
inside admission 110-111
need of care in constructing 161-162
pounding in 433
reduces steam clearance 162
Vauclain compound 162-165
Planimeter 310-312
Power—definition of term 271
calculations 303-312
Possible cause for wrecks.....460-463
Pressure—absolute 267
back 267
boiler or gauge.....267
condenser 268
initial 267
mean effective (M. E. P.).....267-280
mean forward 284
terminal 268
total, or absolute 268
Primary considerations in locomotive design....444-445
Pyle electric head light 406-418
dynamo 410
instructions for applying 406-410
instructions for operating.....410-416
suggestions 417-419
the lamp 412

R

Ratio of expansion 268
Release 48

Repairing cylinder packing 436-438
Richardson balanced valve.....126-128
balance plate 127
packing strip 126-128
Rods—proper time to key up.....432
side—keying up.....434-435

S

Schenectady cross compound.....373-388
carrying water 383
compound to simple 382
drifting 383-386
examination of.....386-388
intercepting valve 374-376
lubrication 382
operation 377-380
smoke arch and cylinder saddle.....373-375
starting compound 380-382
steam distribution 375-377
Sea water—boiling point of.....9-10
composition of 9
point of saturation 9
Setting up wedges 431-432
Shoes and wedges 430-431
Snow—weight per cubic foot.....4
Steam—chest and cylinder casting (sectional).....43
(plan) 44
clearance 270
consumption per H. P. per hour.....275-285-295
efficiency 272-273
maximum theoretical duty of.....272
weight of per cubic foot.....289-290
wire drawn 268

Starting a train 439-442
Strength of various metals.....457-458
Stresses on pedestal jaws 428-430
Suggestions to young runners.....442-443

T

Table of train speeds.....443
principal alloys 457
Tandem Compound 351-368
American Locomotive Co's.....352-365
Baldwin 365-369
principal defects of.....352
testing (illustrated) 357-365
Temperature of water 3
Terminal pressure 267
Thermodynamics—first law of.....271
Theoretical expansion curve.....295-298
Traction power of a locomotive.....450
calculations of 450-452-455
formulæ for cross compound.....451
single expansion 452
Vauclain compound 451-453
Train resistance—(calculating) 445
due to acceleration 450
curve 449-455
grade 449
gravity 447-448
velocity 445-447-448

V

Vacuum 268-270
line 279-280

Valve—Alfree system 181-192
Allen Richardson 128-132
American balanced 165-176
semi-plug 112-116
balanced 116-192
balanced slide 148-180
D slide 43
area of wearing surface 129-130
functions of 43-44
objections to 108
Farrer's valve 146-147
Goodspeed's 132-138
Haley's slide 138-142
Jack Wilson high pressure.....117-126
piston valve 108-115
positions of during stroke.....46-51
Richardson balanced 126-128
travel of 48-49-54
types of for locomotives.....108-192
Watkey's valve 144-145
Valve gear—direct or indirect.....51
Baker-Pilliod 94-102
Joy's 103-106
rules for guidance regarding.....112
Stephenson link 51-54-58
Walschaert 58-84
Valve setting 215-240
adjustments for lead 229-232
adjusting eccentric blades 226-229
adjusting points of cut-off.....232-235
equalizing points of cut-off.....235-238
how to find lap, or lead.....226
locating dead centers 217-224

Valve setting—Continued.
locating eccentrics 216-232
locating port marks 216-217
locating release and exhaust closure.....238-239
maximum port opening 239
travel of valve 239
minimum port opening 239
travel of valve 239
Valve travel 278
Vauclain compound 315-400
by pass valves 325-326
cross-head 324
cylinder and valve chest castings.....318-321
improved actions of boiler.....340-342
operation of 327-332
objects in design of 318-338-340
piston 324-325
repairs on 333-335
rules for testing 342-345
setting valves of 323
suggestions for running 326
tractive force of 338
water relief valves 327
valves and valve motion 320-323

W

Walschaert valve gear 58
angularity of main rod 240
break-downs 243-251
broken combination lever.....248-251
cylinder head 250
eccentric 244-246
piston, or rod 249

Walschaert valve gear—Continued.
radius bar 246
side rod 243-244
suspension bar 246-247
valve stem 247
details of construction 66-71
adjustment 80-84
diagrams of 78-79
distribution of steam with 62
ease of handling 64
economy of 59
general description of 58-59
location of link fulcrum 70
lifter 70
method of laying out 72-74
necessity of correct design 62
source of motion 59-62-71
for lead 80
testing correctness of adjustment 241-242
Watkey's valve 144-145
double exhaust port 145
false seat of 144
Water—boiling point of 15
buoyancy of 14
composition of (fresh) 2
(sea) 6
compressibility of 12
expansion of by heat 2
cold 4
incompressibility of 12
pressure of, due to head 10-11-12
specific gravity of 5
specific heat of 13

Water—Continued.
useful figures on 13-14
volume of at various temperatures.....2-3-5
weight of cu. foot at 39°.....3-4
212° 3-4
1 cylinder inch at 62°.....4
fresh compared to sea water.....5
Wedges—proper time to set up.....432
setting up 430-432
Work—definition of term.....271-272
negative 309
unit of 271

Z

Zero—absolute 270
Zeuner diagram 204-209

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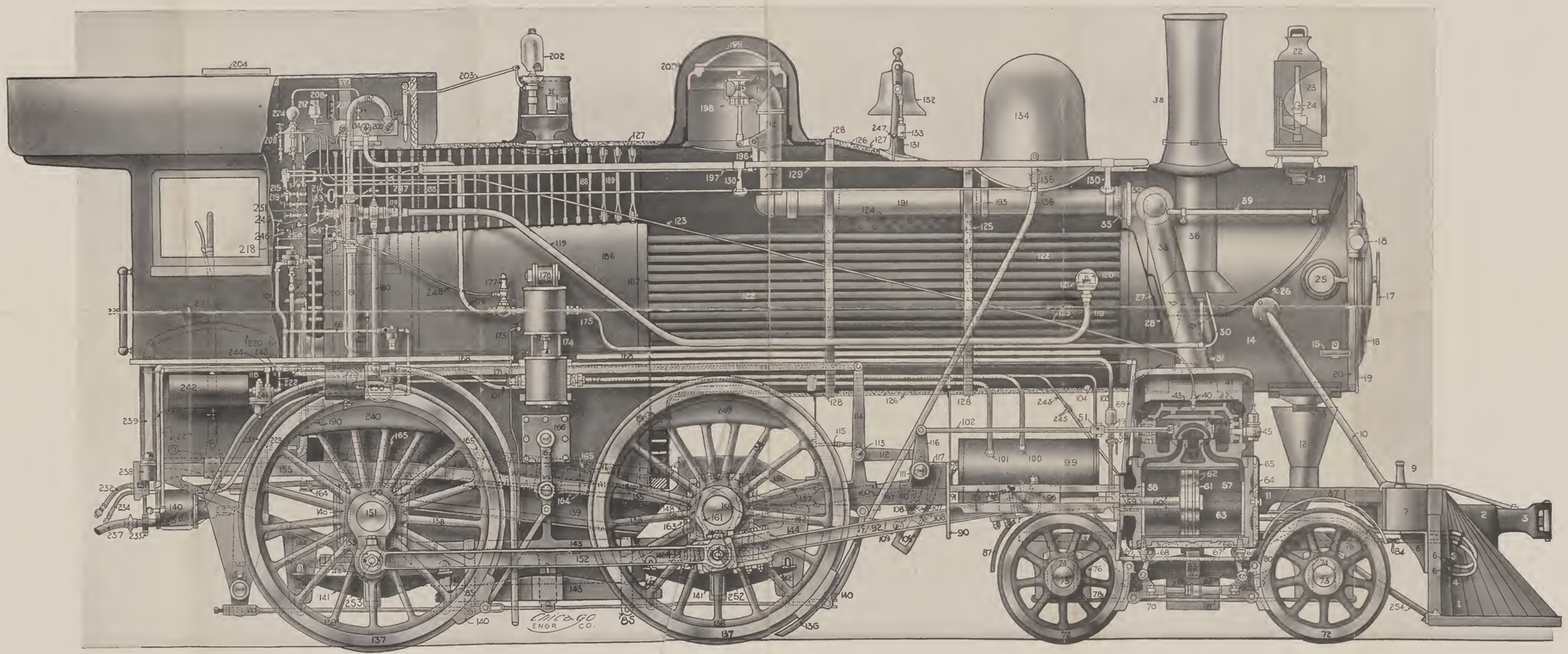

1 Pilot.
2 Draw Bar Plate.
3 Coupler.
4 Air Signal Hose.
5 Air Brake Hose.
6 Hose Hangers.
7 Buffer Beam.
8 Pilot Bracket.
9 Flagstaff.
10 Arch Brace.
11 Front Frame.
12 Cinder Chute.
13 Cinder Chute Side.
14 Extension Front.
15 Headlight Step.
16 Signal Lamp.
17 Number Plate.
18 Smoke Arch Door.
19 Smoke Arch Front.
20 Smoke Arch Ring.
21 Headlight Bracket.
22 Headlight Case.
23 Headlight Reflector.
24 Headlight Burner.
25 Cleaning Door.
26 Netting.
27 Deflector Plate.
28 Deflector Plate Adjuster.
29 Air Pump Exhaust Pipe.
30 Blower.
31 Nozzle Stand.
32 Nozzle Tip.
33 Steam Pipe. (2)
34 T or Nigger Head.
35 Dry Pipe Joint.
36 Petticoat or Draft Pipe.
37 Stack Base.
38 Smoke Stack.
39 Arch Hand Rail.
40 Oil Pipe Plug.
41 Cylinder Saddle.
42 Steam Chest Casing Cover.
43 Steam Chest Cover.
44 Steam Chest.
45 Relief Valve.
46 Balance Plate.
47 Balance Slide Valve.
48 Valve Yoke.
49 Valve Stem.
50 Valve Stem Packing.
51 Valve Stem Connection.
52 Valve Seat.
53 Bridges.
54 Exhaust Port.
55 Front Train Line Cock.
56 Steam Ports.
57 Cylinder.
58 Back Cylinder-Head.
59 Piston Packing.
60 Piston Rod.
61 Piston Head.
62 Piston Packing Rings.
63 Truck Center Casting.
64 Front Cylinder-Head.
65 Cylinder-Head Casing.
66 Cylinder Lagging.
67 Cylinder Casing.
68 Cylinder Cocks.
69 Cylinder Cocks Rigging.
70 Engine Trucks.
71 Engine Truck Wheel.
72 Engine Truck Tire.
73 Engine Truck Axle.
74 Engine Truck Brass.
75 Engine Truck Box.
76 Engine Truck Pedestal.
77 Engine Truck Frame.
78 Engine Truck Pedestal Brace.
79 Engine Truck Frame Brace.
80 Engine Truck Equalizer.
81 Engine Truck Spring Hanger.
82 Engine Truck Spring.
83 Engine Truck Spring Band.
84 Front Signal Line Cock.
85 Safety Hanger.
86 Truck Brake.
87 Wheel Guard.
88 Air Signal Pipe.
89 Guides.
90 Guide Yoke.
91 Guide Block.
92 Main Rod.
93 Main Rod Front Strap.
94 Key.
95 Crosshead Pin.
96 Crosshead.
97 Front Frame.
98 Air Drum Bracket.
99 Air Drum.
100 Pump Connection.
101 Train Pipe Connection from Main Reservoir.
102 Valve Stem Rod.
103 Train Pipe.
104 Wash Out Plugs.
105 Link.
106 Suspension Stud.
107 Link Block Pin.
108 Link Block.
109 Eccentric Connection, Back Up.
110 Eccentric Connection, Go Ahead.
111 Link Hanger.
112 Tumbling Shaft Arm.
113 Tumbling Shaft.
114 Tumbling Shaft Lever.
115 Counter Balance Spring and Rig.
116 Rocker.
117 Rocker Box.
118 Reach Rod.
119 Branch Pipe.
120 Check Valve Case.
121 Check Valve.
122 Flues.
123 Oil Pipe.
124 Horizontal Boiler Seam.
125 Circumferential Seam.
126 Boiler Lagging.
127 Boiler Jacket.
128 Jacket Bands.
129 Hand Rail.
130 Hand Rail Bracket.
131 Bell Stand.
132 Bell.
133 Air Bell Ringer.
134 Sand Box.
135 Sand Box Lever.
136 Sand Pipe.
137 Driving Wheel Tire.
138 Driving Wheel Centers.
139 Ash Pan.
140 Driver Brakes.
141 Driver Springs.
142 Driver Spring Hangers.
143 Driver Spring Equalizers.
144 Driver Spring Hanger Brace.
145 Lower Rail of Frame.
146 Pedestal Brace.
147 Driving Box Shoe.
148 Driving Box Wedge.
149 Wedge Bolt.
150 Driving Box.
151 Driving Axle.
152 Side or Parallel Rod.
153 Rod Bush.
154 Main Rod Connection.
155 Main Frame.
156 Frame Brace.
157 Frame Splice.
158 Go Ahead Eccentric.
159 Back Up Eccentric Rod.
160 Go Ahead Eccentric Rod.
161 Go Ahead Eccentric Strap.
162 Back Up Eccentric Rod.
163 Back Up Eccentric Strap.
164 Grate Shaking Rig.
165 Rocking Grates.
166 Expansion Pad.
167 Expansion Link.
168 Running Board.
169 Air Cylinder Brake Pump.
170 Steam Cylinder Brake Pump.
171 Air Strainer.
172 Delivery to Drum.
173 Drip Cock.
174 Pump Piston Packing.
175 Pump Exhaust Connection.
176 Pump Steam Connection.
177 Governor.
178 Pump Valve Case.
179 Injector.
180 Injector Overflow.
181 Water Pipe.
182 Steam Pipe.
183 Steam Valve.
184 Primer.
185 Water Valve.
186 Fire Box.
187 Tube Sheet.
188 Radial Stay Bolts.
189 Sling Stay.
190 Stay Bolts.
191 Dry Pipe.
192 Stand Pipe.
193 Dry Pipe Hangers.
194 Throttle Pipe.
195 Throttle Valve.
196 Throttle Bell Crank.
197 Throttle Stem.
198 Dome.
199 Dome Cap.
200 Dome Casing.
201 Safety Valves.
202 Chime Whistles.
203 Whistle Rig.
204 Ventilator.
205 Cab.
206 Air Pump Lubricator.
207 Air Gauge.
208 Steam Gauge.
209 Steam Turret.
210 Injector Throttle.
211 Blower Cock.
212 Gauge Lamp.
213 Signal Whistle.
214 Air Pump Throttle.
215 Throttle Lever.
216 Sand Lever.
217 Reverse Lever.
218 Engineer's Brake Valve
219 Gauge Cocks.
220 Quadrant.
221 Cut Out Valve.
222 Fire Door.
223 Cylinder Cock Lever.
224 Sight-Feed Lubricator.
225 Oil Can Shelf.
226 Hand Hold.
227 Shake Lever Stub.
228 Ash Pan Damper Handle.
229 Whistle Signal Valve.
230 Brake Valve Reservoir.
231 Train Pipe.
232 Train Pipe Hose.
233 Signal Pipe.
234 Signal Pipe Hose.
235 Feed Pipe Hanger.
236 Feed Pipe.
237 Feed Pipe Hose.
238 Tail Piece of Frame.
239 Cab Bracket.
240 Counter Balance Weight.
241 Engine Brake Triple Valve.
242 Engine Brake Auxiliary.
243 Connection to Truck Brake Cylinder.
244 Driver Brake Cut Out Cock.
245 Truck Brake Cut Out Cock.
246 Bell Ringer Valve.
247 Air Pipe to Bell Ringer.
248 Air Pipe to Governor.
249 Main Reservoir Connection to Air Gauge.
250 Train Line Connection to Air Gauge.
251 Glass Water Gauge.
252 Main Crank Pin.
253 Back Crank Pin.
254 Pilot Brace.

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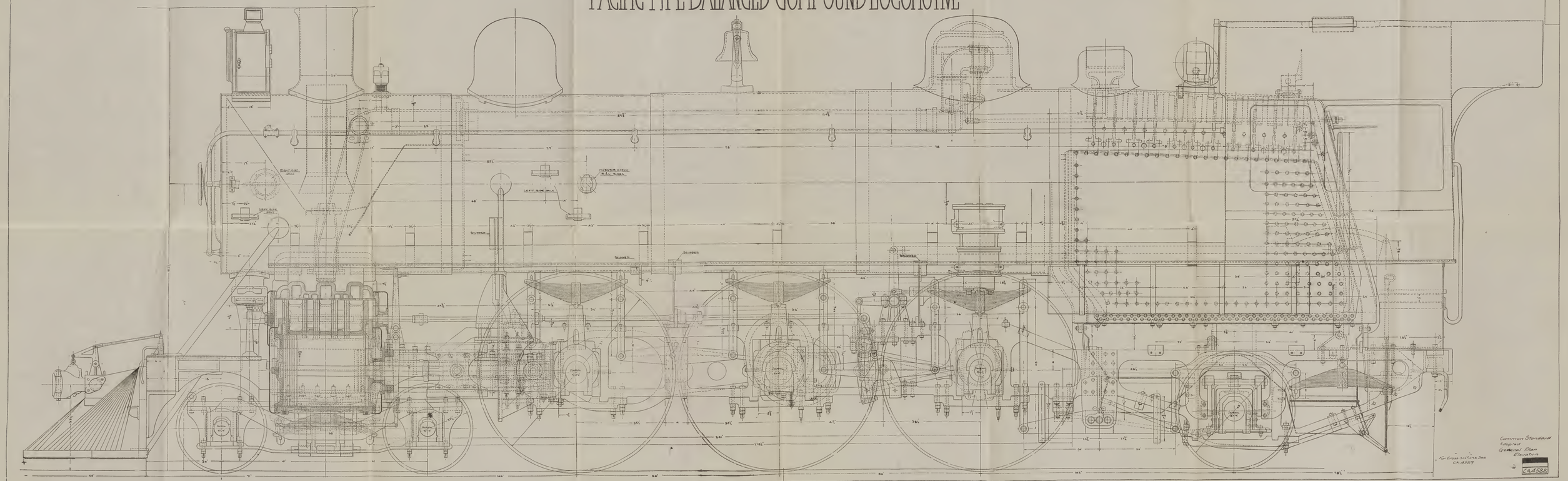
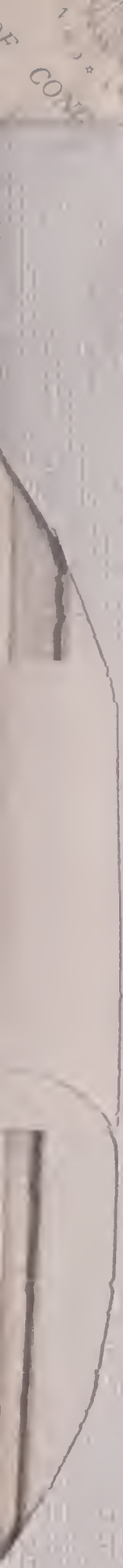
PACIFIC TYPE BALANCED COMPOUND LOCOMOTIVE



Common Standard
Adopted
General Plan
Elevation

For Cross sections See
C.A. 1587

C.A. 1588



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